# PERFORMANCE EVALUATION OF AIR-COOLED HEAT EXCHANGERS

by

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This thesis is prepared in partial fulfilment of the requirement for the degree of Masters of Engineering at the University of Stellenbosch.



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June 2000

# DECLARATION

I, Thabo Makhema, the undersigned, hereby declare that the work contained in this thesis is my own original work and has not previously, in its entirety or in part, been submitted at any other university for a degree.

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Date:

# ABSTRACT

The main objective of this study to determine the air-side heat transfer and pressure drop performance characteristics of commercially available round and elliptical fined tubes. A computer code to compare the performance of industrial forced and induced draught air-cooled heat exchangers (henceforth referred to as ACHE's) commonly found in the petro-chemical industry is also developed. The comparison is extended to include both round and elliptical finned tubes.

From the tests conducted, it is found that there is generally an increase in the heat transfer parameter with a decrease in the fin pitch. The decrease in the fin pitch however also results in an increase in the pressure drop across the tube bundle. The performance of the tubes is compared with round reference tubes having circular or plate fins for which performance correlations are available in the literature. The fan power required by an induced draught air-cooled heat exchanger is found to be higher than that required by a forced draught heat exchanger rejecting the same amount of heat.

# OPSOMMING

Die hoofdoel van hierdie studie is om die werksverrigting van industriële geforseerde trek en geinduseerde trek lugverkoelde warmteruilers (LVWR) soos algemeen aangetref in die petrochemiese nywerheid te vergelyk. Warmteruiler bundels word gewoonlik opgebou uit ronde vinbuise. Die werksverrigting van laasgenoemde word vergelyk met die werksverrigting van warmteruiler bundels wat gebruik maak van elliptiese vinbuise.

Die werksverrigting van 'n reeks kommersieël verkrygbare ronde en elliptiese vinbuise word deur middel van toetsing bepaal. In die algemeen word 'n toename in die warmteoordrag-parameter waargeneem met 'n afname in die vinsteek. 'n Toename in die vinsteek gaan egter gepaard met 'n toename in die drukverlies oor die bundel. Die werksverrigting van hierdie buise word vergelyk met bestaande korrelasies vir die werksverrigting van ronde verwysingsbuise wat gebruik maak van ronde of plaatvinne. Daar is bevind dat die drywing wat benodig word deur 'n waaier vir 'n geinduseerde trek lugverkoelde warmteruiler, heelwat hoër is as in die geval van 'n geforseerde trek lugverkoelde warmteruiler, vir dieselfde hoeveelheid hitte verwydering.

# ACKNOWLEDGEMENTS

I would like to express my gratitude to the following:

1. SASOL for their financial assistance in the form of a bursary and grant to the university.

2. Prof. D.G. Kröger, my study supervisor, for his encouragement, guidance and motivation in difficult times. The practical insight into engineering and his constant insistence on working hard in this study is greatly appreciated.

3. All co-workers for their contributions, especially Kobus Zietsman for his generally helpful nature and for his help with the assembly of the test apparatus.

4. Finally, to my family, especially Basetsana Makhema, for their constant patience and love.

# CONTENTS

## APPENDICES

# LIST OF SYMBOLS

| | |
|---|---|
| A | Area, $m^2$ |
| a | Constant |
| b | Coefficient, or constant, or as defined by equation (5.3.3) |
| C | Heat capacity rate $mc_p$, W/K, or $C_{min}$ / $C_{max}$ |
| $c_p$ | Specific heat at constant pressure, J/kgK |
| $c_v$ | Specific heat at constant volume, J/kgK |
| DALR | Dry adiabatic lapse rate, K/m |
| d | Diameter, m |
| $d_e$ | Equivalent or hydraulic diameter, m |
| Ey | Characteristic pressure drop parameter, $m^{-2}$ |
| e | Effectiveness |
| F | Fan, or correction factor |
| f | Friction factor, or fin |
| G | Mass velocity, kg/ $sm^2$ |
| g | Gravitational acceleration, $m/s^2$ |
| H | Height, m |
| h | Heat transfer coefficient, $W/m^2K$ |
| K | Loss coefficient |
| k | Thermal conductivity, W/mK |
| L | Length, m or mm |
| LMTD | Logarithmic mean temperature difference, °C |
| m | Mass flow rate, kg/s |
| N | Revolutions per minute, $minute^{-1}$, or NTU |
| NTU | Number of transfer units, $UA/C_{min}$ |
| Ny | Characteristic heat transfer parameter, $m^{-1}$ |
| P | Pitch, mm, or power, W |
| p | Pressure, $N/m^2$ or Pa |
| Q | Heat transfer rate, W |
| R | Gas constant, J/kgK, or thermal resistance, $m^2K/W$ |
| Ry | Characteristic flow parameter, $m^{-1}$ |
| T | Temperature, °C or K |
| t | Thickness, mm |

| | |
|---|---|
| U | Overall heat transfer coefficient, $W/m^2K$ |
| V | Volume flow rate, $m^3/s$ or volume, $m^3$ |
| v | Velocity, m/s |
| Y | Defined by equation (3.3.4) |
| z | Elevation, m |

## Greek Symbols

| | |
|---|---|
| $\alpha_e$ | Kinetic energy coefficient |
| $\gamma$ | $c_p/c_v$ |
| $\Delta$ | Differential |
| $\mu$ | Dynamic viscosity, kg/ms |
| $\nu$ | Kinematic viscosity, $m^2/s$ |
| $\xi$ | Temperature lapse rate, K/m |
| $\rho$ | Density, $kg/m^3$ |
| $\sigma$ | Area ratio |
| $\phi$ | Expansion factor defined by equation (3.3.3), or as defined by equation (5.3.4) |

## Dimensionless Groups

| | |
|---|---|
| Eu | Euler number, $\Delta p/ (\rho v^2)$ |
| Nu | Nusselt number, hL/ k for a plate, or hd / k for a tube |
| Pr | Prandtl number, $\mu c_p/ k$ |
| Re | Reynolds number, $\rho vL/ \mu$ for a plate, or a $\rho vd/ \mu$ for a tube |

## Subscripts

| | |
|---|---|
| a | Air, or based on air side area |
| atm | Atmospheric |
| av | Mixture of dry air and water vapour |
| c | Casing, or contraction, or contact |
| do | Downstream |
| db | Drybulb |
| e | Energy, or effective, or equivalent, or expansion |
| F | Fan |

| | |
|---|---|
| Fr | Fan reference |
| Fs | Fan static |
| f | Fin, or fluid |
| fr | Frontal or face |
| g | Ground |
| he | Heat exchanger |
| i | Inlet, or inside |
| iso | Isothermal |
| lm | Logarithmic mean |
| m | Mean, or mixture |
| max | Maximum |
| min | Minimum |
| n | Nozzle |
| o | Outlet, or outside |
| p | Process fluid, or passes |
| pl | Plenum chamber |
| r | Root, or reference |
| rec | Recovery |
| s | Support |
| si | Inlet shroud |
| $\Delta T$ | Constant temperature difference |
| T | Temperature |
| t | Total, or tube, or blade tip |
| tr | Tube row |
| ts | Tube cross-section, or tower support |
| tus | Windtunnel upstream cross-section |
| up | Upstream |
| w | Water |
| wb | Wetbulb |

## Abbreviation

| | |
|---|---|
| ACHE | Air-cooled heat exchanger |
| no. | Number |

# Chapter 1

# INTRODUCTION

SASOL is at present updating the SASOL Specification on Air-Cooled Heat Exchangers and is considering the possibility of including elliptical finned tubes as a design option in its specification. In order to achieve this, more information regarding the performance characteristics of these types of finned tubes is required.

The present study is involved with the evaluation of the thermal-flow performance characteristics of commercially available round and elliptical finned tubes, with a view to comparing their performance in an air-cooled heat exchanger.

The main aims and objectives of this report are to:

- Determine the characteristics of the different finned tubes. This includes the thermal as well as the pressure drop characteristics on the air side.
- Compare the performance of forced and induced draught ACHE's incorporating the different finned tubes tested.

In order to obtain the performance characteristics contained in this report, use was made of an existing code test tunnel at the University of Stellenbosch. Published correlations on heat transfer and pressure drop applicable to finned tubes are also used to extend the scope of this study.

This report begins by firstly describing typical air-cooled heat exchangers commonly used in the petro-chemical industry; namely the forced draught as well as the induced draught air-cooled heat exchangers. The governing energy and draught equations are then derived. The performance comparison of the two types of air-cooled heat exchangers incorporating round finned tubes for which performance correlations are available from the literature is also presented. The report then goes on to describe the different types of finned surfaces that are tested

as well as the test facility used. A description of the test procedure used as well as the format for the presentation of data is given.

The thermal-flow performance characteristics are evaluated for different finned tubes and the results are presented. The effect of fin pitch on the heat transfer as well as the pressure drop is investigated. The effect of varying the water flowrates in the tube on the heat transfer is also examined. For proprietary reasons, geometric and material details of certain tubes are not specified and their characteristics are referred to by a code.

The thermal-flow performance of round reference tubes having circular or plate fins and for which performance correlations are available in the literature are compared with the thermal performance of the commercial tubes tested.

Subsequently, the performance of forced and induced draught air-cooled heat exchangers incorporating the different finned tube bundles is compared. This is achieved by simultaneously solving the two governing equations, namely the energy and draught equations. Results are compared and conclusions are drawn.

# Chapter 2

# AIR-COOLED HEAT EXCHANGERS

## 2.1 Introduction

In view of rising water costs and due to an increased consciousness of environmental problems, power stations and petro-chemical plants are increasingly using water-independent air-cooled cooling systems.

In the petro-chemical industry, two types of mechanical draught air-cooled heat exchangers (henceforth referred to as ACHE's) are commonly found. These are of the **forced draught** type as shown in figure 2.1 where the air is blown through the bundles and the **induced draught** configuration as shown in figure 2.2 whereby the air is drawn through the bundles. The basic design of such a system consists of the heat exchanger (made up of bundles of finned tubes) supported in a metal and/or concrete structure with an axial flow fan creating an air flow through the bundles. The structure supports the heat exchanger at a sufficient elevation above ground level to allow the necessary volume of air to enter at a reasonable approach velocity.

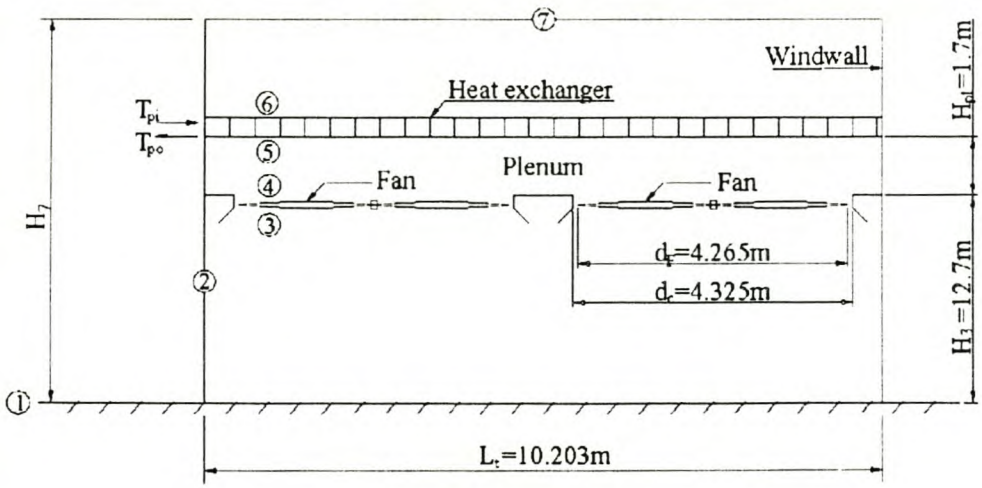



Figure 2.1: Forced draught air-cooled heat exchanger bay.

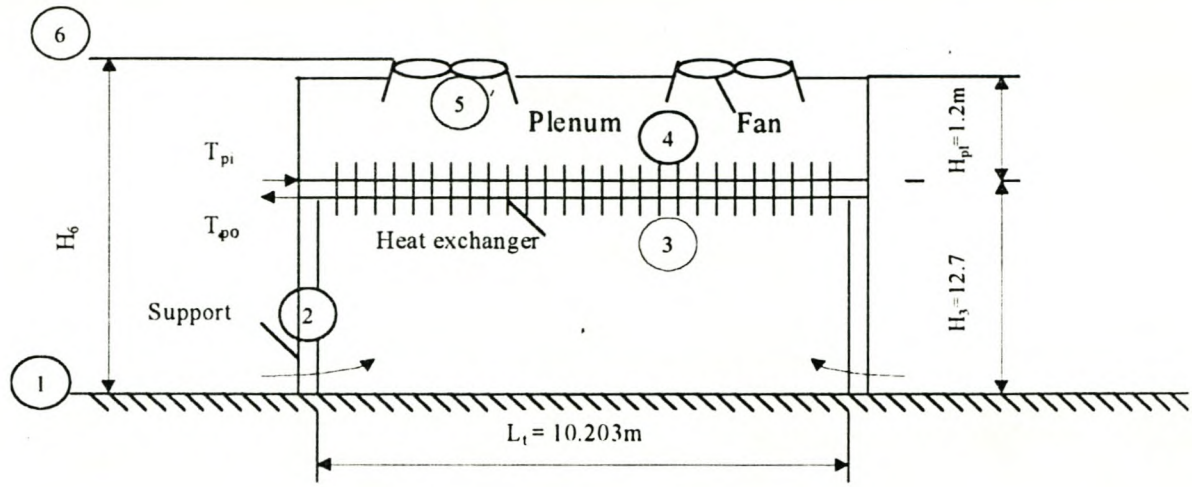



Figure 2.2: Induced draught air-cooled heat exchanger bay.

A brief summary of the main advantages and disadvantages of the two types of layouts follows [91BE1]:

| **Forced draught** | **Induced draught** |
|---|---|
| Advantages: | |
| a) Lower fan shaft power consumption | a) Better distribution of air across the bundle |
| b) Location of fan drives offers better accessibility for maintenance work | b) Relatively high escape of the air from the fan reduces plume recirculation and makes system less susceptible to crosswinds |
| c) Fan drives not exposed to high temperatures making the choice of construction material less critical | c) Bundle is protected in part by plenum from weather conditions |

Disadvantages:

a) Poor air distribution across bundle

b) Low discharge velocity increases the risk of plume recirculation

c) Exposure of the finned surfaces to the atmosphere can affect performance due to wind, rain, hail or solar radiation

a) High fan shaft power consumption

b) Low accessibility of fan components for maintenance work

c) Fan and drive system exposed to the warm air stream making the choice of construction material more critical

## 2.2 Energy and draught equations for a forced and an induced draught ACHE's.

Energy and draught equations for both a forced as well as an induced draught ACHEs will be discussed. These two equations need to be satisfied simultaneously to evaluate the correct air mass flow rate and heat transfer rate. They take into account all flow obstacles and the heat exchanger layout.

### 2.2.1 The energy equations for a forced draught ACHE

The energy equation represents the amount of heat that is transferred to the air stream flowing across the tube bundles from the process fluid flowing inside the tubes. This can be expressed as

$$Q_a = m_a c_{pa}(T_{a6} - T_{a5}) = m_p c_{pp}(T_{pi} - T_{po}) \qquad (2.2.1)$$

where the subscript, p, refers to the process fluid.

The heat transfer rate may also be expressed in terms of the effectiveness of the heat exchanger i.e.

$$Q = eC_{min}(T_{pi} - T_{as}) \tag{2.2.2}$$

where $C_{min}$ is the smaller of $m_a c_{pa}$ and $m_p c_{pp}$ and the effectiveness, e, depends on the geometry and flow patterns of the fluids through the heat exchanger.

Where the air and process fluid mass flow rates are known, it is customary to employ the NTU-effectiveness method in evaluating heat exchanger performance. The effectiveness is defined as

$$e = \frac{Q}{Q_{max}} = \frac{C_{min}\Delta T_{min}}{C_{min}\Delta T_{max}} = \frac{\Delta T_{min}}{\Delta T_{max}} \tag{2.2.3}$$

where

$\Delta T_{min}$ = the temperature difference of fluid having a $C_{min}$

$\Delta T_{max}$ = the maximum temperature difference in the heat exchanger

The effectiveness of numerous practical arrangements is found in various sources in the literature [84KA1], [81SH1] and [91AS1].

### 2.2.2 The draught equation for a forced draught ACHE

In practice the draught for an ACHE is sometimes obtained by simply matching the fan performance curve and the flow characteristics through the heat exchanger bundles only, and evaluating the thermophysical properties of the air at ambient ground level conditions. In some cases, this may give useful approximate values but because of increased competitiveness and high system costs, a more detailed analysis is adopted.

The fan chosen must be able to overcome a series of flow resistances and hence deliver a cooling air flow rate that will guarantee the desired heat transfer rate.

Considering figure 2.1, stagnant ambient air at 1 far from the heat exchanger accelerates and flows across the heat exchanger supports at 2 before reaching the fan at section 3, where upstream obstacles such as structural supports or a screen or mesh guard may be located. After leaving the fan at 4 where further downstream obstacles may be located, the flow experiences losses in the plenum before entering the heat exchanger bundle at 5 and exiting at 6. A windwall of height, $H_w$, is installed to reduce the recirculation of hot plume air exiting at 7.

External to the heat exchanger there is however also a decrease in the ambient pressure with the height of the ACHE. In a 24-hour cycle, the temperature near the surface of the earth may vary significantly. This is due to temperature inversions at night at certain locations and a dry adiabatic lapse rate (DALR) of approximately 0.00975 K/m during the day [98KR1].

Using the DALR, the temperature at any elevation, z, can be written as:

$$T_{az} = T_{a1} - 0.00975z \tag{2.2.4}$$

The pressure gradient in a gravity field is given by

$$dp / dz = - \rho_a g \tag{2.2.5}$$

The air density may be expressed through the perfect gas law as

$$\rho_a = p_a / RT_a \tag{2.2.6}$$

where R = 287.08 J/kgK

Substitute equation (2.2.4) and (2.2.6) into equation (2.2.5) and integrate between points 1 and 7 to obtain the following pressure difference far from the ACHE

$$(p_{a1} - p_{a7}) = (p_{a1} - p_{a6}) + (p_{a6} - p_{a7})$$

$$\approx p_{a1}[1 - (1 - 0.00975\ H_6 / T_{a1})^{3.5}]$$

$$+ p_{a6}\{1 - \{1 - 0.00975\ (H_7 - H_6) / T_{a1}\}^{3.5}] \qquad (2.2.7)$$

where the ambient air temperature at elevation 6 is assumed to be approximately equal to $T_{a1}$. Although the air temperature distribution near ground level generally deviates considerably from the DALR, the error introduced by this assumption in equation (2.2.7) is small for relatively high heat exchangers.

The pressure change across a flow obstacle can be expressed in terms of the loss coefficient of the particular obstacle

$$\Delta p = K\rho_a / 2v_a^2 = K\rho_a / 2(m_a / A)^2 \qquad (2.2.8)$$

The difference in pressure between 1 at ground level and 7 at the outlet of the ACHE may be expressed in terms of the losses experienced by the air stream as it flows through the various obstacles in the ACHE as follows

$$(p_{a1} - p_{a7}) = p_{a1}[1 - (1 - 0.00975H_6/T_{a1})^{3.5}] + K_{ts}(m_a / A_2)^2 / (2\rho_{a1}) + K_{Fsi}(m_a /$$

$$A_c)^2 / (2\rho_{a3}) + K_{up}(m_a / A_e)^2 / (2\rho_{a3}) - (K_{Fs} + K_{rec})(m_a / A_c)^2 / (2\rho_{a3}) + K_{do}(m_a /$$

$$A_e)^2 / (2\rho_{a3}) + K_{he}(m_a / A_{fr})^2 / (2\rho_{a56}) + p_{a6}[1 - (1 - 0.00975(H_7 - H_6)/T_{a6})^{3.5}]$$

$$+ \alpha_{c6}(m_a / A_{fr})^2 / (2\rho_{a6}) \qquad (2.2.9)$$

With the aid of equation (2.2.7), equation (2.2.9) can be written as follows:

$$p_{a6}[\{1 - 0.00975(H_7 - H_6) / T_{a6}\}^{3.5} - \{1 - 0.00975(H_7 - H_6) / T_{a1}\}^{3.5}]$$

$$\approx p_{a1}[\{1 - 0.00975(H_7 - H_6) / T_{a6}\}^{3.5} - \{1 - 0.00975(H_7 - H_6) / T_{a1}\}^{3.5}]$$

(a) (b) (c)

$$= K_{ts}(m_a / A_2)^2 / (2\rho_{a1}) + K_{Fsi}(m_a / A_c)^2 / (2\rho_{a3}) + K_{up}(m_a / A_e)^2 / (2\rho_{a3})$$

(d) (e) (f)

$$- (K_{Fs} + K_{rec})(m_a / A_c)^2 / (2\rho_{a3}) + K_{do}(m_a / A_e)^2 / (2\rho_{a3}) + K_{he}(m_a / A_{fr})^2 / (2\rho_{a56})$$

(g)

$$+ \alpha_{e6}(m_a / A_{fr})^2 / (2\rho_{a3}) \qquad (2.2.10)$$

This is known as the draught equation for the forced draught air-cooled heat exchanger.

It is assumed on the left-hand side of equation (2.2.10) that $p_{a6} \approx p_{a1}$ and $\rho_{a2} \approx \rho_{a1}$, $\rho_{a4} \approx \rho_{a3}$ $\rho_{a7} \approx \rho_{a6}$. It is further assumed that frictional losses between 6 and 7 are negligible and that the kinetic energy factor $\alpha_{e6} \approx \alpha_{e7}$.

The harmonic mean density through the heat exchanger is given by

$$\rho_{a56} = 2p_{a1} / [R(T_{a5} + T_{a6})] \qquad (2.2.11)$$

The different pressure loss coefficients in equation (2.2.10) are:

(a) Heat exchanger support loss support ($K_{ts}$)
(b) Fan shroud inlet loss coefficient ($K_{Fsi}$)
(c) Pressure loss coefficient due to obstacles on the fan suction (upstream) side ($K_{up}$)
(d) The fan static and plenum recovery pressure rise coefficients respectively ($K_{Fs}$, $K_{rec}$)
(e) Pressure loss coefficient due to flow obstacles on the fan discharge (downstream) side ($K_{do}$)
(f) Heat exchanger bundle loss coefficient ($K_{he}$)
(g) Kinetic energy velocity distribution correction factor ($\alpha_{e6}$)

### 2.2.3 The energy and draught equations for an induced draught ACHE.

By following the above procedure, the energy and draught equations can be deduced for the induced draught ACHE shown in figure 2.2

The heat transfer rate is

$$Q = m_a c_{pa}(T_{a4} - T_{a3}) = m_p c_{pp}(T_{pi} - T_{po}) \qquad (2.2.12)$$

or

$$Q = eC_{min}(T_{pi} - T_{a3}) \qquad (2.2.13)$$

where

$$T_{a3} = T_{a1} - 0.00975H_3 - v_{a3}^2 / (2c_{pa}) \approx T_{a1} - 0.00975H_3 \qquad (2.2.14)$$

In an induced draught heat exchanger the air flow through the heat exchanger is usually relatively uniform and upstream turbulence is low.

The draught equation is given by

$$p_{a1}[\{1 - 0.00975(H_6 - H_4) / T_{a4}\}^{3.5} - \{1 - 0.00975(H_6 - H_4) / T_{a1}\}^{3.5}]$$

$$= K_{ts}(m_a / A_2)^2 / (2\rho_{a1}) + K_{he}(m_a / A_{fr})^2 / (2\rho_{a34}) + K_{pl}(m_a / A_c)^2 / (2\rho_{a4}) +$$

$$\alpha_{e6}(m_a / A_c)^2 / (2\rho_{a6}) - K_{Fs}(m_a / A_c)^2 / (2\rho_{a4}) + K_{up}(m_a / A_e)^2 / (2\rho_{a4}) + K_{do}$$

$$(m_a / A_e)^2 / (2\rho_{a4}) \qquad (2.2.15)$$

A performance evaluation of a forced draught ACHE is done in appendix F and a comparison is made with an induced draught ACHE. The results are presented in table F.1 (appendix F).

# Chapter 3

# HEAT TRANSFER SURFACES

## 3.1 Introduction

The most expensive and most critical component of any air-cooled heat exchanger is the heat transfer surface area. The heat exchanger may consist of bundles of one or more rows of finned tubes. The heat transfer from the inside process fluid to the air is influenced by, among others, the following variables

(a) The temperature difference between the fluid and the air.
(b) The design and surface arrangement of the heat exchanger.
(c) The thermophysical properties of the fluid both inside and outside of the tubes.

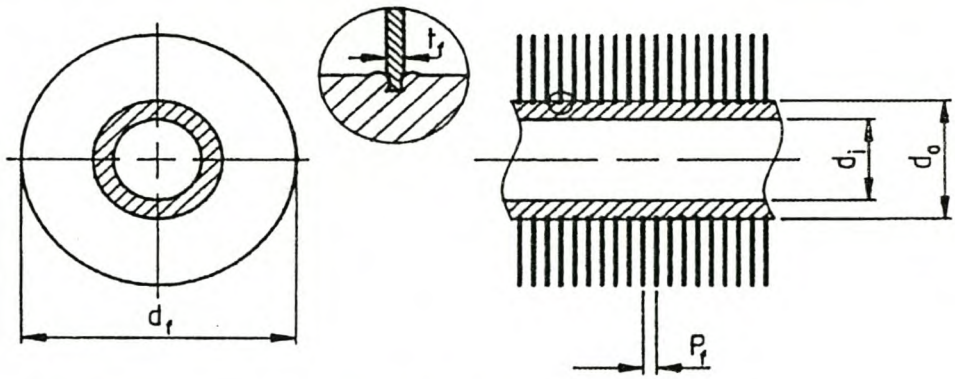
According to M. Abu Madi et al. [98AB1], the heat transfer and pressure loss correlations used on the internal tube surfaces (process fluid side) in most cases provide adequate accuracy. Since the air-side thermal resistance is usually significantly higher than that on the inside of the tube, small errors in predicting the air-side heat transfer performance lead to corresponding errors in predicting the overall thermal performance of heat exchangers. The accuracy of a heat exchanger model may, therefore, be judged by the availability of reliable air-side heat transfers and pressure loss data.

## 3.2 Finned surfaces

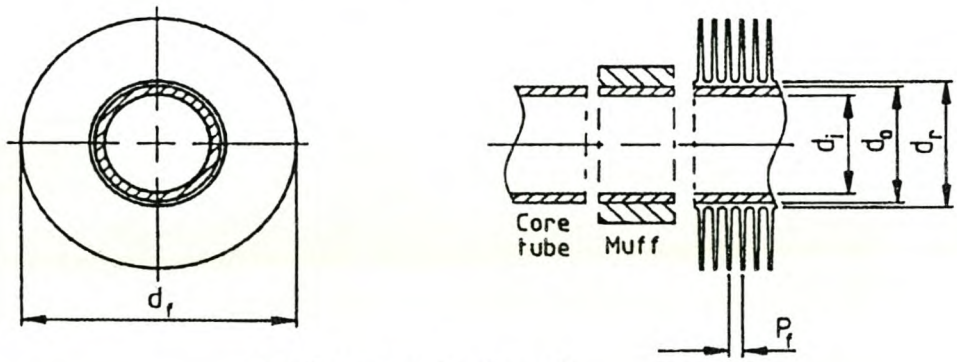
Finned tubes may be round, elliptical, flattened or otherwise streamlined to reduce the flow resistance on the air-side. Kröger [98KR1] illustrates different types and shapes of fins commonly used.

This report is restricted only to the performance evaluation of round and elliptical finned tubes.

Figure 3.1(a) shows a helically wound aluminium G-finned tube. The core tube is provided with a groove into which the fin is rolled, whereafter the groove is peened back against the sides of the fin material. Where corrosion is a major consideration, an extruded bimetallic E-fin tube as shown in figure 3.1(b) is recommended. The finned surface is obtained by plastically deforming an outer aluminium muff onto an internal steel tube during a rolling process. The fins can also be extruded from materials other than aluminium.




**(a) Type G-fin.**




**(b) Extruded fin, Type E.**

Figure 3.1: Grooved and extruded finned tubes

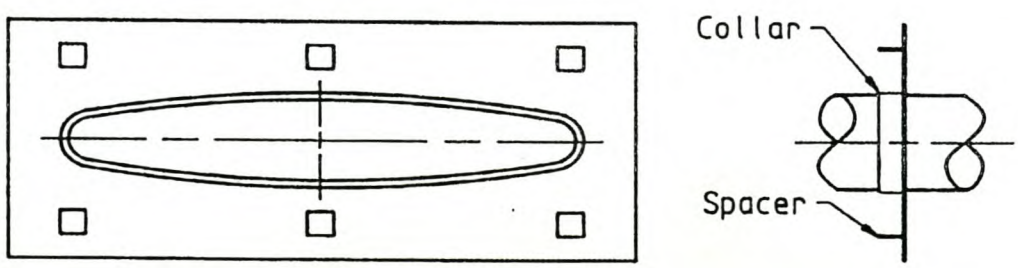



Figure 3.2: Elliptical tube with rectangular plate fin.

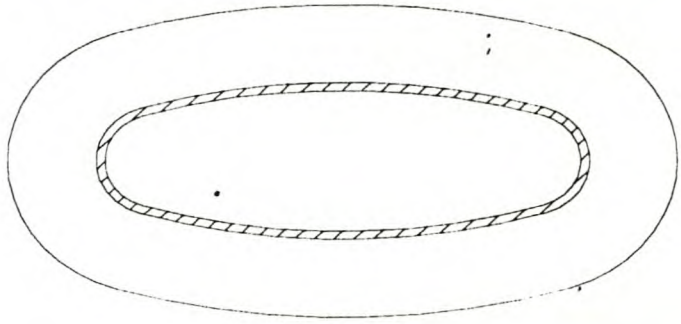


Figure 3.3: Elliptical tube with wrap-on fin (Galvanised).

An elliptical steel tube with a rectangular steel-plate fin and wrap-on fin is shown in figures 3.2 and 3.3 respectively. The finned tube is galvanised after assembly to ensure good thermal contact between fin and tube.

## 3.3 Test facilities and procedure

The Stellenbosch University's code test windtunnel is used to determine the performance characteristics of the extended surfaces. A windtunnel designed specifically for this purpose is shown in figure 3.4.

A radial fan 8 draws air uniformly through the rounded inlet section, where its wet- and drybulb temperature is measured, and then across the heat exchanger bundle 1 which is heated by a fluid flowing inside the tubes. The static pressure difference is measured across the bundle at points located in the duct wall 3. Depending on the type of bundle to be tested, care should be taken that the outlet pressure tap is in a position where it will not be influenced by flow distortions immediately after the bundle. After the heat exchanger, the air passes through an insulated connecting section 2 and two sets of air mixers 4, followed by a venturi in which a sampling tube is located.

The air discharged from the heat exchanger may have a non-uniform temperature distribution together with a non-uniform velocity distribution. The most accurate means of measuring the mean temperature of the air stream under these conditions, is to introduce air mixers and then to sample the

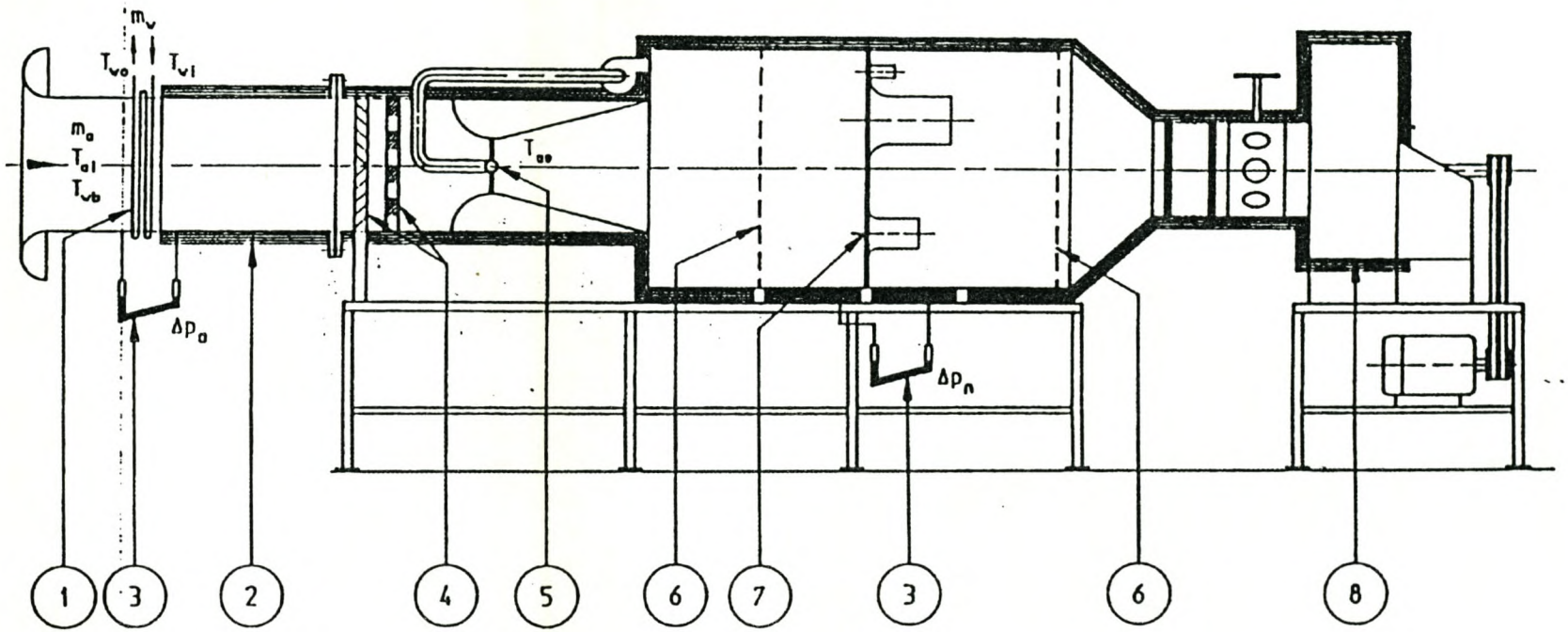



Figure 3.4: Code test windtunnel.

stream at a number of points. Air mixers may consist of a series of vanes arranged to divide the air flow into many small streams that are diverted across each other. The venturi arrangement after the mixers tends to minimize the non-uniformity of the air-stream velocity. The sampling tube 5 permits the withdrawal of air from numerous points across the venturi throat and conveys it to a convenient location where the mean dry- and wetbulb temperature may be measured.

Warm water, steam or some other heated fluid is passed through the tubes. Where warm water is employed, a uniform distribution of flow must be ensured through each of the tube passes. This can be achieved by the correct design of the manifolds. Air-water counterflow conditions are normally preferred.

### 3.3.1 Evaluation of air mass flow rate

The air flow rate is determined by measuring the pressure drop across one or more elliptical nozzles mounted in a plate 7 between perforated plates 6. The corresponding mass flow rate is given by

$$m_a = C_n \phi_g Y A_n (2\rho_n \Delta p_n)^{0.5} \qquad (3.3.1)$$

The nozzle coefficient of discharge, $C_n$, is a function of the nozzle Reynolds number. For $30000 < Re_n < 100000$,

$$C_n = 0.954803 + 6.37817 \times 10^{-7} Re_n - 4.65394 \times 10^{-12} Re_n^2 + 1.33514 \times 10^{-17} Re_n^3 \qquad (3.3.2a)$$

For $100000 < Re_n < 350000$

$$C_n = 0.9758 + 1.08 \times 10^{-7} Re_n - 1.6 \times 10^{-13} Re_n^2 \qquad (3.3.2b)$$

and for $Re_n > 350000$, $C_n = 0.944$

The gas expansion factor $\phi_g$ may be approximated by the following relation:

$$\phi_g = 1 - 3\,\Delta p_n / (4\, p_{up}\, c_p / c_v) \qquad (3.3.3)$$

where $c_p / c_v = 1.4$ for air and $p_{up}$ is the upstream pressure.

For a compressible fluid, it can be shown that the approach velocity factor is approximately

$$Y = 1 + 0.5\,(A_n / A_{tus})^2 + 2\,(A_n / A_{tus})^2\, \Delta p_n / (p_{up}\, c_p / c_v) \qquad (3.3.4)$$

Equation (3.3.1) neglects thermal expansion or contraction of the nozzle.

## 3.4 Presentation of data

Different methods for presenting the heat transfer and pressure drop characteristics of finned tubes are found in the literature. Generally the heat transfer is presented in the form of the dimensionless Nusselt number, i.e.

$$Nu = h d_e / k = a_1\, \mathrm{Re}^{b_1}\, \mathrm{Pr}^{0.333} \qquad (3.4.1)$$

where $d_e$ is some equivalent diameter. Similarly, the pressure differential may be represented by the dimensionless Euler number

$$Eu = \Delta p / \rho v^2 = \rho \Delta p / G^2 \qquad (3.4.2)$$

or a loss coefficient

$$K = 2\Delta p_t / \rho v^2 = 2\Delta p_t / G^2 \qquad (3.4.3)$$

where $\Delta p_t$ is the change in the total pressure across the heat exchanger. The mass velocity may be based on the minimum flow area through the heat exchanger, or the frontal area of the heat exchanger.

A method for presenting the experimentally obtained performance data of finned tubes which has particular merit in the case of industrial finned-tube heat exchangers adopted for the thesis is the one evaluated by Kern [80KE1] and presented by Kröger [86KR1] in a modified form. This method is used to present the performance characteristics of the evaluated tube bundles.

Both the Nusselt number and the Reynolds number contain an equivalent or hydraulic diameter. Because of the relatively arbitrary nature or the definition of this quantity for finned surfaces, different definitions are found in the literature. This may often lead to confusion and makes any comparison of performance characteristics of different types of finned surfaces meaningless. In the absence of the equivalent diameter, equation (3.4.1) may be written as

$$h/k = a_2 Ry^{b_2} \Pr^{0.333} \tag{3.4.4}$$

where

$$Ry = G_{fr}/\mu = m/(\mu A_{fr}) \tag{3.4.5}$$

is known as the characteristic flow parameter.

The effective finned surface area and the heat exchanger frontal area play a significant role in comparing and optimising heat exchangers. These parameters are introduced into equation (3.4.4) to give the following relation:

$$Ny_h = he_f A/\left(kA_{fr} \Pr^{0.333}\right) = a_{Ny_h} Ry^{b_{Ny_h}} \tag{3.4.6}$$

where $Ny_h$ is known as the characteristic heat transfer parameter.

For most cases of industrial finned surfaces that have been performance tested, the actual fin-side heat transfer coefficient, h, and the finned surface effectiveness, $e_f$, is of no particular interest. Henceforth the use of an effec-

tive heat transfer coefficient based on the air-side surface area is preferred to define a corresponding heat transfer parameter

$$h_{ae}A_a = \left( \frac{1}{h_a e_f A_a} + \sum_n \frac{R_n}{A_n} \right)^{-1} = \left( \frac{F_T \Delta T_{lm}}{Q} - \frac{1}{h_w A_w} \right)^{-1} \tag{3.4.7}$$

where $h_{ae}$ is the effective air-side heat transfer coefficient.

The logarithmic mean temperature difference is given by

$$\Delta T_{lm} = \frac{(T_{wo} - T_{ai}) - (T_{wi} - T_{ao})}{\ln[(T_{wo} - T_{ai})/(T_{wi} - T_{ao})]} \tag{3.4.8}$$

The experimentally determined value of the heat transfer rate, $Q_a$, is determined by calculating the heat transfer on the air-side of the heat exchanger bundle as follows

$$Q_a = m_a c_{pam} (T_{ao} - T_{ai}) \tag{3.4.9}$$

According to Roetzel [84RO1], a temperature correction factor can in general be expressed as

$$F_T = 1 - \sum_{i=1}^{4} \sum_{k=1}^{4} a_{i,k} (1 - \phi_{cf})^k \sin\left( 2i \arctan \frac{\phi_h}{\phi_c} \right) \tag{3.4.10}$$

where the dimensionless temperature changes of the two streams may be respectively defined as

$$\phi_h = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}} \tag{3.4.11}$$

and

$$\phi_c = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}} \tag{3.4.12}$$

For a counterflow case, a dimensionless mean temperature difference can be expressed in terms of the logarithmic mean temperature difference as follows

$$\phi_{cf} = \frac{\Delta T_{lm}}{T_{hi} - T_{ci}} \tag{3.4.13}$$

Kröger [98KR1] gives values of $a_{i,k}$ for ten different heat exchanger geometries.

The water-side heat transfer coefficient is determined by the following equation as proposed by Gnielinski [75GN1]

$$h_w = \frac{Nu_w d_e}{k_w} = \frac{(f_D/8)(\mathrm{Re}_w - 1000)\,\mathrm{Pr}\left[1 + (d_e/L)^{0.67}\right]}{1 + 12.7(f_D/8)^{0.5}(\mathrm{Pr}^{0.67} - 1)}\left(\frac{d_e}{k_w}\right) \tag{3.4.14}$$

For turbulent flow inside smooth tubes, the Darcy friction factor, $f_D$, is determined by the following equation as proposed by Filonenko [54FI1]

$$f_D = (1.82 \log_{10}\mathrm{Re}_w - 1.64)^{-2} \tag{3.4.15}$$

The pressure loss coefficient based on the total pressure difference across the heat exchanger and the mean density can be defined as

$$K_{heiso} = \frac{\Delta p_t}{\frac{1}{2}\rho_m v_m^2} = a_k Ry^{b_k} \tag{3.4.16}$$

For non-isothermal flow conditions [98KR1], the resultant loss coefficient is

$$K_{he} = K_{heiso} + \frac{2}{\sigma^2}\left(\frac{\rho_i - \rho_o}{\rho_i + \rho_o}\right) = a_k Ry^{b_k} + \frac{2}{\sigma^2}\left(\frac{\rho_i - \rho_o}{\rho_i + \rho_o}\right) \tag{3.4.17}$$

where $\sigma$ is the ratio of the minimum air-side flow area between tubes and the corresponding frontal area. Thermophysical properties are evaluated at the arithmetic mean temperature $T_m = (T_i + T_o) / 2$.

As mentioned previously this method of data presentation enables one to readily compare the performance of different heat exchangers.

# Chapter 4

# PRESENTATION OF EXPERIMENTAL DATA AND RESULTS

## 4.1 Introduction

In order to effectively design a finned tube air-cooled heat exchanger (ACHE) knowledge of the heat transfer as well as the pressure drop characteristics of the finned tubes to be used is required. These performance characteristics enable the designer to predict the heat transfer and pressure drop across a particular heat exchanger bundle. The performance characteristic results presented in this chapter are determined using the code test windtunnel described in section 3.3. A method for the presentation of the experimental heat exchanger data as presented in section 3.4 is employed. Tubes of different geometries and from different manufactures are tested and results are presented and subsequently discussed.

## 4.2 Effect of fin pitch and the water mass flowrate on the thermal-flow performance of a heat exchanger bundle.

Four different elliptical finned tube bundles (AE-type) having fin pitches of 2.2, 2.3, 2.4 and 2.5 mm are evaluated. The aim of the investigation is to determine the effect of fin pitch on the air-side performance characteristics of the tubes.

The tube specifications, test data and thermal-flow performance results are tabulated and presented in appendix A from table A.1.1 through to table A.4.5. These results are further presented graphically from figure A.1.1 through to figure A.4.2. The effects of fin pitch on the heat transfer parameter and pressure coefficient is graphically shown from figure A.5 through to figure A.7.

As expected, there is an increase in the heat transfer parameter with a decrease in the fin pitch, as shown in figure A.5. The decrease in fin pitch however also results in a higher pressure drop across the bundle as is shown in figures A.6 and A.7. The non-isothermal

pressure loss coefficient conditions, which the tube is likely to experience under normal operation, is always higher than for a corresponding isothermal case.

The effect of varying the water mass flowrate on the heat transfer parameter, everything else being held constant, is also determined and the results are tabulated and presented in tables A.5.1 (a) through to A.5.2 (b) for tubes AE-2.4. The results are further presented graphically in figure A.8. From the results presented, it is shown that varying the water mass flowrate has essentially no effect on the heat transfer parameter, Ny. This result is to be expected and confirms the reliability of the test results for Ny which are shown to be independent of the water flow rate.

A sample calculation for determining the heat transfer characteristics for a single pass air-cooled heat exchanger bundle with elliptical finned tubes is presented in appendix B.

**4.3 Thermal-flow performance of the C-2.5 and C-4.3 elliptical finned tubes.**

The heat transfer and pressure drop characteristics of the C-2.5 and C-4.3 elliptical finned tubes (AE-type) are determined experimentally. The following configurations are considered.

(a) The performance characteristics of a heat exchanger bundle consisting of one row of elliptical finned tubes having a 2.5 mm fin pitch (C-2.5) and a heat exchanger consisting of one row of elliptical finned tubes having a 4.3 mm fin pitch (C-4.3) respectively, are determined experimentally. The tube specifications, data and results of these tests are tabulated and presented in appendix C from tables C.1.1 through to C.2.5

(b) A heat exchanger bundle consisting of two elliptical finned tube rows (upstream fin pitch 4.3 mm, i.e. C-4.3, downstream fin pitch 2.5 mm, i.e. C-2.5) is tested. The resultant data and test results are tabulated and presented in appendix C from tables C.3.1 (a) through to C.3.4

The results of both (a) and (b) are further presented graphically from figures C.1 through to C.3. It is observed that there is a consistent behaviour where the two-row heat exchanger has the highest heat transfer parameter as well as the pressure drop coefficient (isothermal

and non-isothermal). The C-4.3 tube bundle exhibits the lowest heat transfer parameter as well as the lowest pressure drop coefficient (isothermal and non-isothermal). This is consistent with the findings observed in appendix A concerning the effect of fin pitch on the heat transfer characteristics.

### 4.4 Thermal-flow performance of elliptical finned tubes (type AE).

The heat transfer and pressure drop characteristics of different type AE elliptical finned tubes are determined experimentally and the following configurations are considered.

(a) The performance characteristics of a heat exchanger bundle consisting of two elliptical finned tube rows having type DE turbulators and a 2.5 mm fin pitch (AE-2.5 DE) upstream and downstream respectively are determined experimentally. This is called a double row test. The tube specifications, data and results of this test are tabulated and presented in appendix D from tables D.1.1 through to D.1.5. The results are further presented graphically in figures D.1.1 and D.1.2

(b) The performance characteristics of a heat exchanger bundle consisting of two elliptical finned tube rows having type DE turbulators and a 4 mm fin pitch (AE-4.0 DE) upstream and downstream respectively are determined experimentally. The tube specifications, data and results of this double row test are tabulated and presented in appendix D from tables D.2.1 through to D.2.5. The results are further presented graphically in figures D.2.1 and D.2.2.

(c) The performance characteristics of a heat exchanger bundle consisting of two elliptical finned tube rows having type DA turbulators and a 4.3 mm fin pitch (AE-4.3 DA) upstream and downstream respectively are determined experimentally. The tube specifications, data and results of this double row test are tabulated and presented in appendix D from tables D.3.1 through to D.3.5. The results are further presented graphically in figures D.3.1 and D.3.2.

(d) The performance characteristics of a heat exchanger bundle consisting of two elliptical finned tube rows having type DA turbulators and a 4.5 mm fin pitch (AE-4.5 DA) upstream and downstream respectively are determined experimentally. The tube specifications, data and results of this double row test are tabulated and presented in

appendix D from tables D.4.1 through to D.4.5. The results are further presented graphically in figures D.4.1 and D.4.2.

(e) The performance characteristics of a heat exchanger bundle of two elliptical finned tube rows (upstream fin pitch 4 mm, i.e. type DE turbulator AE-4.0 DE tube, downstream fin pitch 2.5 mm, i.e. type DE turbulator AE-2.5DE tube) are determined experimentally. The resultant data and test results are tabulated and presented in appendix D from tables D.5.1 (a) through to D.5.4. The results are further represented graphically in figures D.5.1 and D.5.2.

(f) The performance characteristics of a heat exchanger bundle of two elliptical finned tube rows (upstream fin pitch 4.3 mm, i.e. type DA turbulator AE-4.3 DA tube, downstream fin pitch 2.5 mm, i.e. type DE turbulator AE-2.5DE tube) are determined experimentally. The resultant data and test results are tabulated and presented in appendix D from tables D.6.1 (a) through to D.6.4. The results are further represented graphically in figures D.6.1 and D.6.2.

(g) The performance characteristics of a heat exchanger bundle of two elliptical finned tube rows (upstream fin pitch 4.5 mm, i.e. type DA turbulator AE-4.5 DA tube, downstream fin pitch 2.5 mm, i.e. type DE turbulator AE-2.5DE tube) are determined experimentally. The resultant data and test results are tabulated and presented in appendix D from tables D.7.1 (a) through to D.7.4. The results are further represented graphically in figures D.6.1 and D.6.2.

Comparison of the performance characteristics of the different double row as well as the combination tests for the elliptical finned tubes are shown graphically from figures D.8 through to D.13. For the double row test the tube bundle having the 2.5 mm fin pitch (AE-2.5 DE) has the highest heat transfer parameter as well as the highest pressure drop coefficient whereas the 4.5 mm fin pitch (AE-4.5 DA) tube bundle has the lowest performance characteristics. It is observed that though the AE-4.3 DA) tube bundle has a smaller pitch than the 4.3 mm fin pitch (AE-4.3 DA) tube bundle, hence more available fin area for heat transfer, it has lower performance characteristics. This could be attributed to the fact that the fins have different turbulators, which affect their performance differently.

## 4.5 Thermal-flow performance of the ED-, F- and R-finned tubes.

The heat transfer and pressure drop characteristics of the elliptical ED- and F-type finned tubes as well as the round G- and E-type (see figure 3.1) finned tubes are determined experimentally and the following configurations are considered.

(a) The performance characteristics of a heat exchanger bundle consisting of four tube rows having four tube passes, made up of the elliptical ED-2.5L and ED-2.5S finned tubes (similar to one shown in figure 3.3) are determined experimentally. The tube specifications, data and results of these tests are tabulated and presented in appendix E from tables E.1.1 through to E.1.5 and from table E.2.1 through to E.2.5 respectively. These results are further shown graphically in figures E.1 to E.4.

(b) The performance characteristics of a heat exchanger bundle consisting of six tube rows having a three row and three tube passes, made up of the elliptical F-3.0 finned tubes are determined experimentally. The tube specifications, data and results of these tests are tabulated and presented in appendix E from tables E.3.1 through to E.3.5. These results are further shown graphically in figures E.5 and E.6.

(c) The performance characteristics of a heat exchanger bundle consisting of four tube rows having four tube passes, made up of round R-2.4G (G-fin) and R-2.3D (Extruded fin) (19.6mm and 20.86mm inside tube diameters respectively) finned tubes are determined experimentally. The tube specifications, data and results of these tests are tabulated and presented in appendix E from tables E.4.1 through to E.4.5 and from table E.5.1 through to E.5.5 respectively. These results are further shown graphically in figures E.7 to E.10.

Comparison of the heat transfer parameter as well as the pressure drop coefficient of the above-mentioned finned tubes is shown graphically in figures E.11 and E.12 respectively. In figure E.11, it is shown that the tube bundle having the ED-2.5L finned tubes has the highest heat transfer parameter whilst the one for the R-2.4G has the lowest. Scattering of data is seen for the latter from figure E.7. Furthermore, it has lower performance characteristics than for a similar tube, the R-2.3D finned tube.

# Chapter 5

# PERFORMANCE COMPARISON OF DIFFERENT FINNED TUBE BUNDLES

## 5.1 Introduction

Heat exchanger bundles having different finned tubes are installed in a forced draught ACHE (similar to figure 2.1) with the view to comparing their respective thermal-flow performance. The heat exchanger that performs best will be the one requiring the least amount of fan and process fluid pumping power for a given heat rejection rate. The specified heat to be rejected is 5.022 MW at a water flowrate of 100 kg/s given a water velocity of $v_w \approx 2.5$ m/s through the tubes. This water velocity through the reference extruded bimetallic finned tube bundle (see section 5.2), which forms a basis for comparing the thermal-flow performance of the different finned tubes, ensures that no cavitation occurs through the tubes. Reasonably low noise levels need to be maintained and a recommended fan blade tip speed of $v_t = 60$ m/s or less is prescribed.

## 5.2 Determine the thermal-flow performance of a reference extruded finned tube.

To illustrate the procedure for performance evaluation of heat exchanger bundles having different finned tubes, an existing extruded finned round tube (to be referred to as reference tube and shown in figure 5.1) heat exchanger bundle is considered. The thermal-flow performance characteristics of a heat exchanger having such tubes are determined by Kröger [98KR1] and the results obtained are used as a reference value to enable comparison between different heat exchanger bundles.

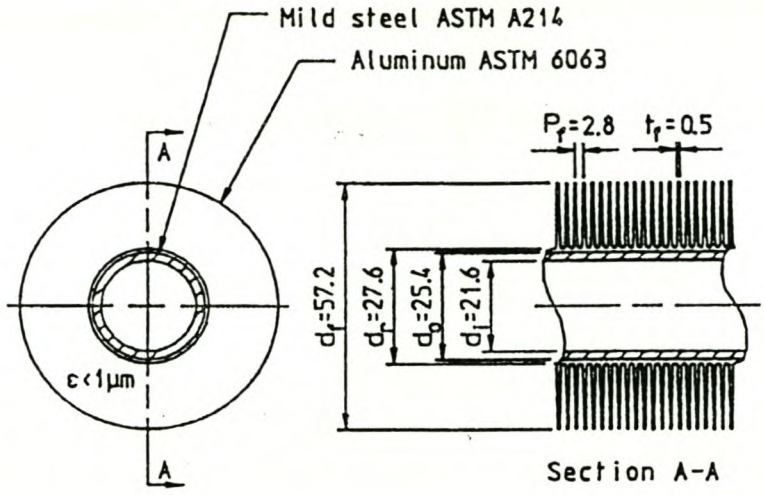



Figure 5.1: Extruded bimetallic round finned tube.

Details of the extruded reference finned tube shown in figure 5.1 are as follows:

| | |
|---|---|
| Fin material | ASTM 6063 aluminium |
| Thermal conductivity | $k_f$ = 204 W/mK |
| Fin diameter | $d_f$ = 57.2 mm |
| Fin root diameter | $d_r$ = 27.6 mm |
| Fin shape | Tapered |
| Fin tip thickness | $t_{ft}$ = 0.25 mm |
| Fin thickness (mean) | $t_f$ = 0.5 mm |
| Fin root thickness | $t_{fr}$ = 0.75 mm |
| Fin pitch | $P_f$ = 2.80 mm |
| Fin surface roughness | $\varepsilon_f$ < 1 μm |
| Tube material | ASTM A214 mild steel |
| Thermal conductivity | $k_t$ = 50 W/mK |
| Tube outside diameter | $d_o$ =25.4 mm |
| Tube inside diameter | $d_i$ = 21.6 mm |
| Mean thermal contact resistance | $R_c = 4 \times 10^{-5}$ $m^2K/W$ |
| | |
| Tube arrangement | Staggered |
| Effective length of finned tube | $L_t$ = 10.203 m |
| Transversal tube pitch | $P_t$ = 58 mm |
| Longitudinal tube pitch | $P_l$ = 50.22 mm |

| | |
|---|---|
| Number of tube rows | $n_r = 4$ |
| Number of tubes per row | $n_{tr} = 55$ |
| Number tube passes | $n_p = 2$ |

The experimentally determined characteristic heat transfer parameter for the bundle is given by

$$Ny = 383.617313 Ry^{0.523761}$$

The isothermal loss coefficient through the bundle is

$$K_{heiso} = 1383.94795 Ry^{-0.332458}$$

A 4.265-m diameter fan is used in this application. The fan blade tip speed is given by

$$v_t = N_F \, d_F \, (\pi/60) \qquad (5.2.1)$$

where

$N_F$ = fan rotational speed, rpm

$d_F$ = fan diameter, m

The maximum allowable fan rotational speed is found using equation (5.2.1)

$$N_F = 60 v_t/(\pi d_F) = 60 \times (60/\pi) \times 1 / 4.265 = 268.68 \text{ rpm}$$

**5.2.1 Evaluating the pressure drop across (inside) the reference tubes.**

In order to be able to determine the required process fluid pumping power, the pressure drop between the inlet to and outlet of a tube is evaluated.

The Reynolds number for the water flowing in the tube is

$$\mathrm{Re}_w = \frac{4m_w}{\pi d_i \mu_w n_{tr}} = \frac{4x100}{\pi x 0.0216 x 4.631 x 10^{-4} x 55 x 2} = 115713.9$$

The flow in the tube is turbulent. The water mass velocity through the tubes is thus

$$G_w = \frac{m_w}{n_t A_{ts}} = \frac{100x4}{55x2x\pi x0.0216^2} = 2480.9 kg/m^2 s$$

The frictional pressure drop may be determined according to the following equation

$$\Delta p_f = f_D \left( \frac{L_t}{d_i} \right) \frac{G_{wi}^2}{2\rho_{wi}} \qquad (5.2.2)$$

For a smooth tube, the friction factor according to equation (3.4.15) is

$f_D = (1.82\log_{10}Re_w - 1.64)^{-2} = (1.82\log_{10}115713.9 - 1.64)^{-2} = 0.017426$

The frictional pressure drop is thus according to equation (5.2.2)

$$\Delta p_f = 0.017426 \left( \frac{10.203}{0.0216} \right) \frac{2480.9^2}{2x983.22} = 25763.5 Pa$$

For the particular tube layout, the area ratio for the entering water stream is

$\sigma = A_c / A_{fr} = \pi \times 21.6^2 / (4 \times 58 \times 50.22) = 0.125803$

The jet contraction ratio for round tubes [98KR1] is

$\sigma_c = 0.61375 + 0.13318\sigma - 0.26095\sigma^2 + 0.51146\sigma^3$

$= 0.61375 + 0.13318 \times 0.125803 - 0.26095 \times 0.125803^2 + 0.51146 \times 0.125803^3$

$= 0.62739$

For turbulent flow, the inlet contraction loss coefficient may be approximated [50KA1] by

$K_c = 1 - 2 / \sigma_c + 1 / \sigma_c^2 = 1 - 2 / 0.62739 + 1 / 0.62739^2 = 0.3527$

The corresponding static pressure drop at the inlet to the tube is

$\Delta p_i = 0.5\rho_w v_w^2[(1 - \sigma^2) + K_c] = 0.5 \times 983.22 \times 2.5^2[(1 - 0.125803^2) + 0.3527] = 4107.63$ Pa

The outlet expansion coefficient is approximated by

$K_e = (1 - \sigma^2) = (1 - 0.125803)^2 = 0.7642$

The corresponding static pressure drop (recovery) at the outlet of the tube is

$\Delta p_e = 0.5\rho_w v_w^2[K_e - (1 - \sigma^2)] = 0.5 \times 983.22 \times 2.5^2[0.7642 - (1 - 0.125803^2)] = -675.88$ Pa

Because the inlet ($\Delta p_i$) and outlet ($\Delta p_e$) pressure drops are relatively small, they will be neglected in all further process fluid pressure drop evaluation.

## 5.3 Heat transfer results

Numerous heat transfer and pressure drop correlations for flow through bundles of round finned tubes have been reported in the literature [78MC1],[98AB1]. In the following section, a theoretical prediction of the heat transfer and pressure drop for staggered circular and plate finned tubes will be presented. These results will be compared with corresponding experimental results.

### 5.3.1 Theoretical and experimental comparison of the heat transfer for staggered circular finned tubes.

Ganguli et al. [85GA1] proposed the following heat transfer correlation for three or more rows of staggered circular finned tubes:

$$Nu = h\, d_r / K = 0.38\, Re^{0.6}\, Pr^{0.333}\, (A/A_r)^{-0.15} \qquad (5.3.1)$$

Figure 5.2 below shows the experimentally determined heat transfer parameter for the reference extruded finned round tube and this is compared with the Ganguli correlation [98KR1]. The experimentally determined values for the R-2.3D finned tube (see table E.5.4b, appendix E) is also plotted. The reference tube is compared to the R-2.3D finned tube as the two are similar and thus should have similar performance characteristics.

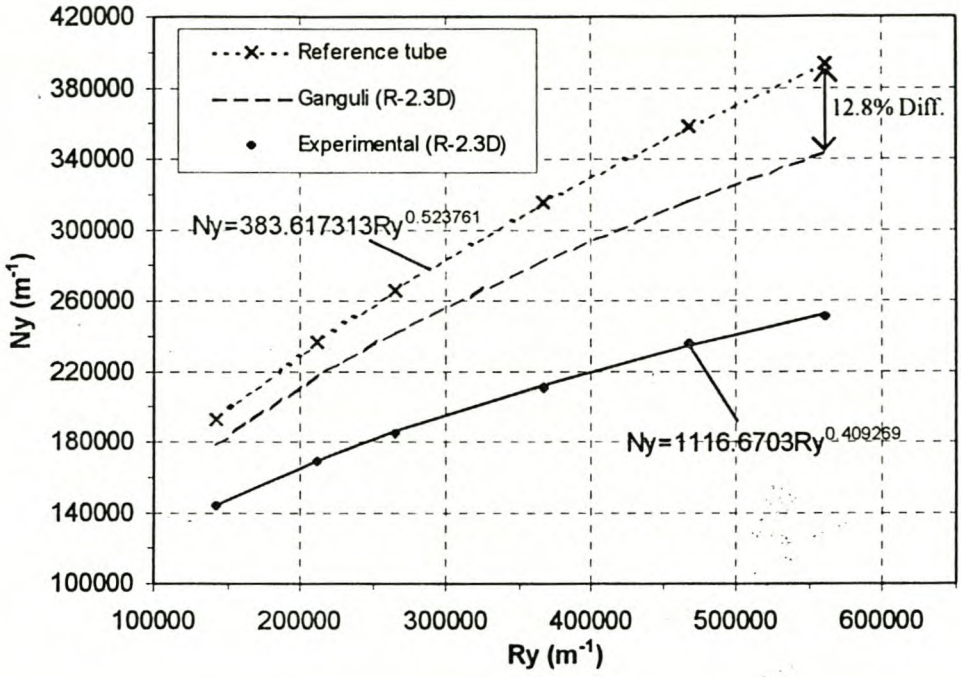



Figure 5.2: Experimental and theoretical prediction of the heat transfer parameter.

It is shown from figure 5.2 that the heat transfer parameter for the tested R-2.3D finned tube bundle (experimental) is well below that predicted by the Ganguli correlation and that of the reference extruded finned tube. Visual inspection of the R-2.3D finned tube showed that the fin made poor contact with the tube resulting in a possible significant thermal contact resistance.

### 5.3.2 Determining the thermal contact resistance for a R-2.3D finned round tube.

The following section evaluates the approximate thermal contact resistance between the fin and tube of the R-2.3D finned round tube assuming the correlation of Ganguli et al. [85GA1] to be applicable.

Details of the R-2.3D finned tube are as follows:

| | |
|---|---|
| Fin material | Aluminium |
| Thermal conductivity | $k_f$ = 204 W/mK |
| Fin diameter | $d_f$ = 57.85 mm |
| Fin thickness | $t_f$ = 0.3 mm |

| | |
|---|---|
| Fin pitch | $P_f = 2.35$ mm |
| Tube material | mild steel |
| Thermal conductivity of tube | $k_t = 50$ W/mK |
| Tube outside diameter | $d_o = 25.4$ mm |
| Tube inside diameter | $d_i = 20.86$ mm |
| Tube arrangement | Staggered |
| Number of tube rows | $n_r = 4$ |
| Number of tubes per row: | |
| effective | $n_{tr} = 7.5$ |
| actual | $n_{tr} = 8$ |
| Transversal tube pitch | $P_t = 64$ mm |
| Longitudinal pitch | $P_l = 60$ mm |
| Length of finned tube (effective) | $L_t = 0.47$ m |
| Frontal area | $A_{fr} = 0.2256$ m$^2$ |

The total effective airside fin surface area is given by

$$A_f = [2(\pi d_f^2/4 - \pi d_o^2/4)]\, L_t n_r n_{tr} / P_f$$

$$= (\pi/2)\,(d_f^2 - d_o^2)\, L_t n_r n_{tr} / P_f$$

$$= \pi/2 \times [(57.85 \times 10^{-3})^2 - (25.4 \times 10^{-3})^2] \times 0.47 \times 4 \times 7.5 / 2.35 \times 10^{-3}$$

$$= 25.461 \text{ m}^2$$

By adding the exposed root area to this value the total airside surface area is found

$$A_a = A_f + \pi d_o (P_f - t_f)\, L_t n_r n_{tr} / P_f$$

$$= 25.461 + \pi \times 25.4 \times 10^{-3}(2.35 \times 10^{-3} - 0.3 \times 10^{-3}) \times 0.47 \times 4 \times 7.5 / 2.35 \times 10^{-3}$$

$$= 26.442 \text{ m}^2$$

The minimum free flow area through the heat exchanger is given by

$A_c = A_{fr} - n_{tr}L_t[d_f t_f + (P_f - t_f) / d_o] / P_f$

$= 0.2256 - 7.5 \times 0.47[57.85 \times 0.3 \times 10^{-6} + (2.35 - 0.3) \times 25.4 \times 10^{-6}] / 2.35 \times 10^{-3}$

$= 0.12146\ m^2$

The total effective water-side surface area is given by

$A_w = n_r n_{tr} \pi d_i L_t = 4 \times 7.5 \times \pi \times 0.02086 \times 0.47 = 0.92402\ m^2$

**Solution**

The following sample calculation is done for test run no.1 (see appendix E, tables 4.1 through to E.4.5) where the following properties are measured/calculated.

| | | |
|---|---|---|
| Dynamic viscosity of air | $\mu_a$ | $= 1.864 \times 10^{-5}$ kg/ms |
| Thermal conductivity of air | $k_a$ | $= 2.66 \times 10^{-2}$ W/mK |
| Prandtl number of air | $Pr_a$ | $= 0.713087$ |
| Air inlet temperature | $T_{ai}$ | $= 27.0579$ °C |
| Air outlet temperature | $T_{ao}$ | $= 39.8052$ °C |
| Water inlet temperature | $T_{wi}$ | $= 59.2123$ °C |
| Water outlet temperature | $T_{wo}$ | $= 55.8337$ °C |
| Temperature correction factor | $F_T$ | $= 0.9984551$ |
| Heat transfer rate | $Q_a$ | $= 30598$ W |

Properties of water stream evaluated at the arithmetic mean temperature,

$$T_{wm} = (T_{wi} + T_{wo})/2 = (332.3623 + 328.9837)/2 = 330.673\ \text{K}$$

Specific heat of water is

$$c_{pw} = 8.15599x10^3 - 2.80627x10T + 5.11283x10^{-2}T^2 - 2.17582x10^{-13}T^6$$

$$= 8.15599x10^3 - 2.80627x10(330.673) + 5.11283x10^{-2}(330.673)^2 - 2.17582x10^{-13}(330.673)^6$$
$$= 4182.562\,J/kgK$$

Dynamic viscosity of water is

$$\mu_w = 2.414x10^{-5}x10^{247.8/(T-140)} = 2.414x10^{-5}x10^{247.8/(330.673-140)} = 4.8122 \text{ x } 10^{-4} \text{ kg/ms}$$

Thermal conductivity of water is found from

$$k_w = -6.14255x10^{-1} + 6.9962x10^{-3}T - 1.01075x10^{-5}T^2 + 4.74737x10^{-12}T^4$$

$$= -6.14255x10^{-1} + 6.9962x10^{-3}(330.673) - 1.01075x10^{-5}(330.673)^2$$

$$+ 4.74737x10^{-12}(330.673)^4 = 0.65076 \text{ W/mK}$$

The Prandtl number of water is then found to be

$$\text{Pr}_w = \frac{\mu_w c_{pw}}{k_w} = \frac{4.8122x10^{-4}x4182.562}{0.65076} = 3.0929$$

We now determine the waterside heat transfer coefficient. Firstly, calculating the waterside Reynolds number.

$$\text{Re}_w = \frac{4m_w}{\pi d_i \mu_w} = \frac{4x2.1877}{\pi x0.02086x4.8122x10^{-4}} = 277485.256$$

It follows for smooth tubes that the friction factor is

$$f_w = (1.82\log_{10}\text{Re}_w - 1.64)^{-2} = (1.82\log_{10}(277485.256) - 1.64)^{-2} = 0.0146331$$

From the Gnielinski equation, with $L_t = 4 \text{ x } 0.47 = 1.88$ m, the water-side heat transfer coefficient is,

$$h_w = \frac{k_w f_w (\text{Re}_w - 1000)x\,\text{Pr}_w\left[1 + (\frac{d_e}{L_t})^{0.67}\right]}{d_e x8\left[1 + 12.7\left(\frac{f_w}{8}\right)^{0.5}(\text{Pr}_w^{0.67} - 1)\right]}$$

$$= \frac{0.65076x0.014633\,1(277485.256-1000)x3.0929\left[1+(\frac{20.86}{1880})^{0.67}\right]}{0.02086x8\left[1+12.7\left(\frac{0.0146331}{8}\right)^{0.5}(3.0929^{0.67}-1)\right]}$$

$$= 31710.92 \text{ W/m}^2\text{K}$$

Furthermore, the logarithmic mean temperature difference (LMTD) is given by

$$\Delta T_{lm} = \frac{(T_{wo}-T_{ai})-(T_{wi}-T_{ao})}{\ln\left[\frac{(T_{wo}-T_{ai})}{(T_{wi}-T_{ao})}\right]}$$

$$= \frac{(55.8337-27.05794)-(59.2123-39.8052)}{\ln\left[\frac{(55.8337-27.05794)}{(59.2123-39.8052)}\right]} = 23.7847K$$

According to equation (5.3.1);

$h_a d_r / k = 0.38\ Re^{0.6} Pr^{0.333} (A_a/A_r)^{-0.15}$

where

$Re = G_c d_r/\mu_a = m_a d_r/(A_c \mu_a) = 2.35856 \times 0.0254 / (0.12146 \times 1.864 \times 10^{-5}) = 26460.71$

therefore

$h_a = (2.66\times10^{-2}/0.0254) \times 0.38 \times 26460.71^{0.6} \times 0.713087^{0.333} \times (26.442 / 0.9815)^{-0.15}$

$= 97.7076 \text{ W/m}^2\text{K}$

For a radial fin of uniform thickness, Schmidt [46SC1] proposes the following equation for determining the fin efficiency

$\eta_f = \tanh(bd_o\phi/2) / (bd_o\phi/2)$ (5.3.2)

where

$b = (2h_a / k_f t_f)^{0.5} = (2 \times 97.7076 / 204 \times 0.3 \times 10^{-3})^{0.5} = 56.50715$ (5.3.3)

and

$\phi = (d_f/d_o - 1)[1 + 0.35\ln(d_f/d_o)]$ (5.3.4)

$= (57.85/25.4 - 1)[1 + 0.35\ln(57.85/25.4)] = 1.64561$

therefore

$\eta_f = \tanh(56.50715 \times 0.0254 \times 1.64561 \times 0.5) / (56.50715 \times 0.0254 \times 1.64561 \times 0.5)$

$= 0.70092$

Surface effectiveness is given as

$e_f = 1 - A_f(1 - \eta_f) / A_a = 1 - 25.461(1 - 0.70092) / 26.442 = 0.712$ (5.3.5)

A characteristic flow parameter, Ry is found using equation (3.4.5)

$Ry = m_a / (A_{fr} \times \mu_a) = 2.35856 / (0.2256 \times 1.864 \times 10^{-5}) = 560869.63\ m^{-1}$

Similarly, a characteristic heat transfer parameter follows from equation (3.4.6)

$Ny = h_a\, e_f\, A_a / k_a\, A_{fr}\, Pr^{0.333}$

$= 97.7076 \times 0.712 \times 26.442 / (2.66 \times 10^{-2} \times 0.2256 \times 0.713087^{0.333}) = 343078.537\ m^{-1}$

Using equation (3.4.7);

$$\left[\frac{1}{h_a e_f A_f} + \sum_n \frac{R_n}{A_n}\right] = \left[\frac{F_T \Delta T_{lm}}{Q} - \frac{1}{h_w A_w}\right]$$

Assuming negligible resistance due to the fin root, the $R_n/A_n$ term includes resistances due to the tube wall as well as the thermal contact resistance at the steel-aluminium interface.

From the above equation,

$$\sum_n \frac{R_n}{A_n} = \left[ \frac{F_T \Delta T_{lm}}{Q} - \frac{1}{h_w A_w} - \frac{1}{h_a e_f A_a} \right]$$

$$= \left[ \frac{0.9984551x23.7847}{30598} - \frac{1}{31710.92x0.92402} - \frac{1}{97.7076x0.712x26.442} \right]$$

$$= 0.000198377 \text{ K/W}$$

$$= \frac{1}{n_r n_{tr} L_t} \left[ \frac{\ln\left(d_o / d_i\right)}{2\pi k_t} + \frac{R_c}{\pi d_o} \right]$$

The mean thermal contact resistance is therefore given by

$R_c = [0.000198377 \text{ x } 4 \text{ x } 7.5 \text{ x } 0.47 - \ln(25.5/20.86) / (2\pi \text{ x } 50)] \pi \text{ x } 0.0254$

$= 1.7219 \text{ x } 10^{-4} \text{ m}^2\text{K/W}$

This value is more than four times higher than the specified thermal contact resistance of the reference extruded finned tube which has a value of $R_c = 4 \text{ x } 10^{-5} \text{ m}^2 \text{ K/W}$.

### 5.3.3 Theoretical and experimental comparison of the isothermal pressure drop for staggered circular finned tubes.

Considering the R-2.3D fin tube, Ganguli et al. [85GA1] propose the following equation for the pressure drop through bundles for staggered circular finned tube:

$Eu = \rho\Delta p/G_c^2 = 2n_r[1 + 2 \exp\{-(P_t - d_f) / (4d_r)\} / \{1 + (P_t - d_f) /d_r\}]$
$[0.021 + 13.6 (d_f - d_r) / Re (P_f - t_f) + 0.25246\{(d_f - d_r) / Re (P_f - t_f)\}^{0.2}]$

This equation is applicable since $2.5 < (d_f - d_r)/[2(P_f - t_f)] = (57.85 - 25.4)/[2(2.35 - 0.3)]$

$= 7.9146 < 12.5$

For the four-row bundle

$$\Delta p = 2 \times 4 \times 19.41845^2 / (1.1276) \times [1 + 2 \exp\{- (64 - 57.85) / (4 \times 25.4)\} / \{1 +$$

$$(64 - 57.85) / 25.4\}] [0.021 + 13.6(57.85 - 25.4) / (26457.1545 \times (2.35 - 0.3)) +$$

$$0.25246\{(57.85 - 25.4) / (26457.1545 \times (2.35 - 0.3))\}^{0.2}] = 581.187 \text{ N/m}^2$$

This value compares with the measured experimental pressure differential of 582 $N/m^2$ (Appendix E, table E.5.3, test run no.1).

The pressure drop was also determined at other flow rates and the results are plotted in figure 5.3 below.

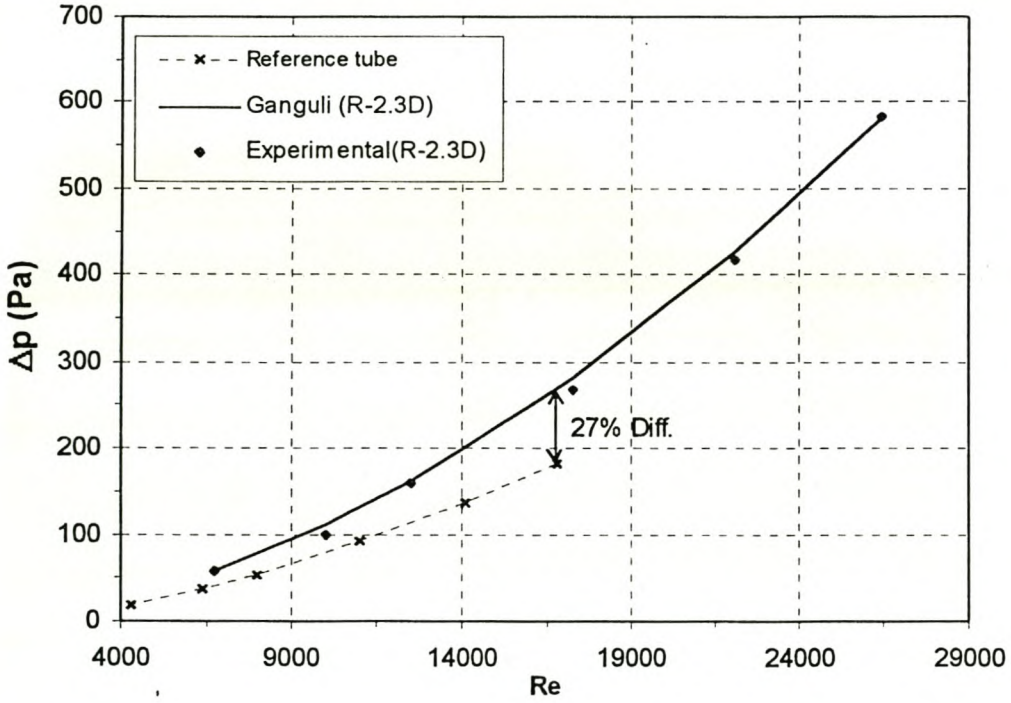



Figure 5.3: Experimental and theoretical prediction of the airside pressure drop.

## 5.4 McQuiston heat transfer correlation for staggered plate-finned round tubes.

In addition to correlations for round tubes with radial fins, correlations are also available for round tubes with plate fins. In the following analysis, the performance of an industrial plate finned heat exchanger will be determined. The application of this type of finned tube in an ACHE will also be compared with the performance of the other finned tubes being studied.

McQuiston [78MC1] presents heat transfer data for staggered plate-finned round tubes in dimensionless form according to a method originally proposed by Colburn [33CO1],

$$j = \frac{h\mathrm{Pr}^{0.67}}{G_c c_p} = \text{function of Re} \qquad (5.4.1)$$

where j is known as the Colburn j-factor.

Details of the heat exchanger whose thermal-flow performance characteristics are to be evaluated using the McQuiston correlation are as follows:

| | |
|---|---|
| Tube arrangement | Staggered |
| Number of tube rows | $n_r = 8$ |
| Number of tubes per row | $n_{tr} = 40$ |
| Transversal tube pitch | $P_t = 20$ |
| Longitudinal tube pitch | $P_l = 20$ |
| Frontal area | $A_{fr} = 0.5808\ m^2$ |
| Tube material | Copper |
| Thermal conductivity of tube | $k_t = 364$ W/mK |
| Tube outside diameter | $d_o = 10$ mm |
| Tube inside diameter | $d_i = 8$ mm |
| Fin material | ASTM 6063 aluminium |
| Thermal conductivity of fin | $k_f = 204$ W/mK |
| Fin thickness | $t_f = 0.24$ mm |
| Fin pitch | $P_f = 3$ mm |
| Flow length through the bundle parallel to the air flow | L = 160 mm |
| Width of bundle perpendicular to air flow | W = 800 mm |

The total effective airside fin surface area is given by

$$A_f = [2\, t_f(L + W) + 2\,(L \times W - n_{tr} n_r \pi d_o^2/4)]\, L_t / P_f$$

$$= [\,2 \times 0.00024(0.16 + 0.8) + 2(0.8 \times 0.16 - 40 \times 8 \times \pi \times 0.01^2/4)]\, 726/3$$

$$= 49.899\ m^2$$

By adding the exposed root area to this value the total airside surface area is found

$$A_a = A_f + A_r = A_f + \pi d_o(P_f - t_f)\, L_t n_r\, n_{tr} / P_f$$

$$= 49.899 + \pi \times 0.01(3 \times 10^{-3} - 0.24 \times 10^{-3}) \times 40 \times 8 \times 0.726 / (3 \times 10^{-3}) = 56.614\ m^2$$

The minimum free flow area through the heat exchanger is given by

$$A_c = n_{tr} L_t [P_t P_f - (d_f - d_r) t_f - P_f d_r] / P_f$$

$$= 40 \times 726[20 \times 3 \times 10^{-6} - (20 - 10) \times 0.24 \times 10^{-6} - 3 \times 10 \times 10^{-6}] / 3$$

$$= 0.26717\ m^2$$

The area ratio is then

$$\sigma = A_c / A_{fr} = 0.26717 / 0.5808 = 0.46$$

The hydraulic diameter is defined by

$$D_h = 4LA_c / A_a = 4 \times 160 \times 0.26717 / 56.614 = 3.02\ mm$$

During the particular performance test to evaluate the heat transfer coefficient, $h_a$, the following thermophysical properties at an air mean temperature of $T_{am}$ = 33.4316 °C (306.5816 K) are

| | |
|---|---|
| Density | $\rho_a = 1.12759$ kg/m$^3$ |
| Specific density | $c_{pa} = 1017.717$ J/kgK |
| Dynamic viscosity | $\mu_a = 1.864 \times 10^{-5}$ kg/ms |
| Thermal conductivity | $k_a = 2.6606 \times 10^{-2}$ J/kgK |
| Prandtl number | $Pr_a = 0.713$ |

For this particular test, an air mass flowrate of $m_a = 2.35857$ kg/s is used.

### 5.4.1 Evaluating the heat transfer parameter.

Two particular parameters in addition to the j factors feature predominantly in heat transfer correlations [78MC1]. The first is the Reynolds number based on the outside tube diameter [73RI1].

$$Re_D = G_c d_o / \mu_a = m_a d_o / (A_c \mu_a) \qquad (5.4.2)$$
$$= 2.35857 \times 0.01 / (0.26717 \times 1.864 \times 10^{-5}) = 4736.04$$

The second is the ratio of total surface area to the outside surface area of the tubes without fins, $A/A_t$, which is expressed by McQuiston as

$$\frac{A}{A_t} = \frac{4}{\pi} \frac{P_l}{D_h} \frac{P_t}{D} \sigma \qquad (5.4.3)$$

$$= 4/\pi \times (20/3.02) \times (20/10) \times 0.46 = 7.7575$$

The following correlating parameter for use with the j factors is obtained

$$JP = Re_D^{-0.4} (A/A_t)^{-0.15} \qquad (5.4.4)$$
$$= 4736.04^{-0.4} (7.7575)^{-0.15} = 0.02491$$

Rich [75RI1] shows that the Reynolds number based on longitudinal tube spacing is important in expressing row effect.

$$Re_b = G_c P_l / \mu_a \qquad (5.4.5)$$
$$= 2.35857 x\, 0.02 / (0.26717 x\, 1.864 x\, 10^{-5}) = 9472.08$$

The j factor for an eight-row tube bundle is given by

$$j_8 = \left[\frac{1-1280 n_r \, \mathrm{Re}_b^{-1.2}}{1-5120\,\mathrm{Re}_b^{-1.2}}\right][0.0014+0.2618(JP)] \qquad (5.4.6)$$

$$= \left[\frac{1-1280x8x9472.08^{-1.2}}{1-5120x9472.08^{-1.2}}\right][0.0014+0.2618x0.02491] = 0.0071704$$

From equation (5.4.1), the heat transfer coefficient is

$h_a = j_8 G_c c_{pa} Pr_a^{-0.67} = 0.0071704 \, x \, 2.35857/0.26717 \, x \, 1017.717 \, x \, 0.713^{-0.67} = 80.809 \ W/m^2K$

The fin efficiency of a continuous plate fin having a staggered tube array using equation (5.3.2) is

$\eta_f = \tanh (bd_o\phi/2) / (bd_o\phi/2)$

where from equation (5.3.3)

$b = (2h_a / k_f t_f)^{0.5} = (2 \, x \, 80.809 / 204 \, x \, 0.24 \, x \, 10^{-3})^{0.5} = 57.455$

and for a staggered tube layout

$d_{fe}/d_r = 2.54(L_1/d_r)(L_2/L_1 - 0.3)^{0.5} = 2.54(10/10)(10/10 - 0.3)^{0.5} = 2.12512$

hence

$\phi = (d_{fe}/d_r - 1)(1 + 0.35\ln(d_{fe}/d_r) = (2.12512 - 1)(1 + 0.35\ln(2.12512)) = 1.421966$

The fin efficiency is thus

$\eta_f = \tanh (57.455 \, x \, 0.01 \, x \, 1.421966 \, x \, 0.5) / (59.972 \, x \, 0.01 \, x \, 1.421966 \, x \, 0.5) = 0.94786$

Surface effectiveness is given as

$e_f = 1 - A_f(1 - \eta_f) / A_a = 1 - 49.899(1 - 0.94786) / 56.614 = 0.95404$

A characteristic flow parameter, Ry, is found using equation (3.4.5)

$Ry = m_a / (A_{fr} \; x \; \mu_a) = 2.35857 / (0.5808 \, x \, 1.864 \, x \, 10^{-5}) = 217859.3756 \; m^{-1}$

Similarly, a characteristic heat transfer parameter is given by

$Ny = h_a \, e_f \, A_a / k_a \, A_{fr} \, Pr^{0.333}$

$= 80.809 \, x \, 0.95404 \, x \, 56.614 / (2.66066 \, x \, 10^{-2} \; x \, 0.5808 \, x \, 0.713^{0.333}) = 316122.684 \; m^{-1}$

The characteristic heat transfer parameter for this bundle was also determined at other air-flow rates and the data may be correlated by the following empirical relation over the range tested:

$Ny = 8 \, x \, 10^{-4} Ry^{1.6329}$

**5.4.2 Evaluating the pressure drop parameter.**

The correlating parameter necessary for the evaluation of the friction data is

$$\frac{R^*}{R} = \frac{A / A_t}{(P_t - d_o)P_f^{-1} + 1} = \frac{7.7575}{(0.02 - 0.01)/0.003 + 1} = 1.7902$$

A parameter to be used to obtain the fanning friction factor, f, is evaluated as follows

$$FP = \mathrm{Re}_D^{-0.25} \left(\frac{R}{R^*}\right)^{0.25} \left[\frac{(P_t - d_o)P_f^{-1}}{4(1 - P_f^{-1} t_f}\right]^{-0.4} \left[\frac{P_t}{2R^*} - 1\right]^{-0.5} \qquad (5.4.7)$$

$$= 4736.04^{-0.25} (1.7902)^{-0.25} \left[\frac{(20 - 10)/3}{4(1 - 0.24/3)}\right]^{-0.4} \left[\frac{20}{1.7902x10} - 1\right]^{-0.5} = 0.31669$$

The fanning friction factor is hence given by

$$f = 4.904\,x\,10^{-3} + 1.382(FP)^2 = 4.904\,x\,10^{-3} + 1.382(0.31669)^2 = 0.14352$$

The approach air velocity is given by

$$v_a = m_a/(\rho_a A_{fr}) = 2.35857/(1.12759\,x\,0.5808) = 3.6014 \text{ m/s}$$

The isothermal heat exchanger bundle loss coefficient is found using equation (3.4.16)

$$K_{he} = \frac{2\rho_a \Delta p_{bundle}}{(m_a / A_{fr})^2}$$

where

$$\Delta p_{bundle} = f_D(L/D_h)\rho_a v_a^2/2 = 4\,x\,0.14352\,x\,(160/3.02)\;x\;1.12759\,x\,3.6014^2\,/2$$
$$= 222.403 \text{ N/m}^2$$

hence

$$K_{he} = \frac{2x1.12759x222.407}{(2.35857/0.5808)^2} = 30.415$$

It is noted that the Darcy friction factor, $f_D$, is $f_D = 4f$.

The loss coefficient for this bundle was also determined at other air flow rates and the data may be correlated by the following empirical equation over the range tested:

$$K_{he} = 12165Ry^{-0.4875}$$

This bundle was further installed in a forced draught ACHE and the thermal-flow performance thereof evaluated. Results are presented in table 5.1 below.

Table 5.1: Performance evaluation of a forced draught ACHE.

| | **Force draught ACHE** |
|---|---|
| Heat rejected, $Q_{rem}$ (MW) | 5.022 |
| Air mass flow rate, $m_a$ (kg/s) | 210.0379 |
| Fan shaft power, $P_F$ (kW) | 20.796 |
| Air outlet temperature, °K | 324.8489 |
| Pressure drop across tube, $\Delta p_f$ (Pa) | 107912.9 |
| Water velocity in tube, $v_w$ (m/s) | 2.86 |

## 5.5 Performance comparison of different finned tubes in a forced draught ACHE.

Table 5.2 below presents the finned tube bundle specifications for the different commercial tubes as tested by the author and then installed in a forced draught ACHE.

Table 5.2: Finned tube bundle geometries installed in a forced ACHE.

| | **R-2.3D** | **F-3.0** | **ED-2.5S** | **ED-2.5L** | **AE-2.5 DE** |
|---|---|---|---|---|---|
| Effective length of finned tube, $L_t$ ($m^2$) | 10.203 | 10.203 | 10.203 | 10.203 | 10.203 |
| Cross-sectional tube inside area, $A_{ts}$ ($m^2$) | $3.418 \times 10^{-4}$ | $2.553 \times 10^{-4}$ | $2.881 \times 10^{-4}$ | $5.994 \times 10^{-4}$ | $1.33 \times 10^{-3}$ |
| Inside/hydraulic diameter of tube, $d_i$ (m) | 0.02086 | 0.01502 | 0.014316 | 0.020848 | 0.025922 |
| Number of tube rows per pass, $n_r$ | 2 | 2 | 2 | 1 | 1 |
| Number of tubes per row, $n_{tr}$ | 55 | 117 | 92 | 72 | 72 |
| Number of tube passes, $n_p$ | 2 | 2 | 2 | 2 | 2 |

From the results obtained in sections 5.2 and 5.3, a performance evaluation of the forced draught ACHE was done and the results are presented in table 5.3 below.

Table 5.3: Performance evaluation of a forced ACHE for different finned tubes.

| | **Reference** | **R-2.3D** | **F-3.0** | **ED-2.5S** | **ED-2.5L** | **AE-2.5DE** |
|---|---|---|---|---|---|---|
| Heat rejected, $Q_{rem}$ (MW) | 5.022 | 5.022 | 5.022 | 5.022 | 5.022 | 5.022 |
| Air mass flow rate, $m_a$ (kg/s) | 258.4607 | 317.7933 | 274.8075 | 243.2662 | 223.312 | 201.171 |
| Fan shaft power, $P_F$ (kW) | 27.5085 | 32.0583 | 33.5807 | 28.2515 | 19.5147 | 17.9184 |
| Fan rotational speed, $N_F$ (rpm) | 248 | 265 | 265 | 250 | 221 | 215 |
| Air outlet temperature, $T_{a6}$ (K) | 320.4046 | 316.7982 | 319.2743 | 321.62 | 323.4227 | 325.8787 |
| Pressure drop across tube, $\Delta p_f$ (Pa) | 25763.5 | 30446.96 | 15465.74 | 22967.61 | 21204.21 | 3617.18 |
| Water velocity in tube, $v_w$ (m/s) | 2.52 | 2.71 | 1.70 | 1.92 | 2.36 | 1.06 |

From the results presented in table 5.3 above, the AE-2.5 DE finned tube heat exchanger performs best as it rejects the specified heat of $Q_{rem}$ = 5.022 MW with the least amount of fan shaft power of $P_f$ = 17.9184 kW. It also has the smallest pressure drop across the tube of $\Delta p_f$ = 3617.18 Pa resulting in the least amount of process fluid pumping power required.

The two ED-type elliptical finned tube heat exchanger bundles perform better than the standard reference round tube heat exchanger bundle. Only the R-2.3D (poor quality tube due to high thermal contact resistance) as well as the F-3.0 elliptical finned tube bundles performs poorly when compared to the standard reference round tube heat exchanger bundle.

# Chapter 6

# CONCLUSIONS

The objective of this thesis was to evaluate the thermal-flow performance characteristics of commercially available elliptical finned tubes, with a view to comparing their performance in forced and induced draught air-cooled heat exchangers.

The first step, in chapter 2, was to consider two types of mechanical draught ACHE's commonly in use in the petro-chemical industry. These are of the forced draught type as well as the induced draught type. The governing equations to evaluate the performance of these two types of ACHE's are employed in a computer performance evaluation program. With this program, it is possible to determine the process fluid outlet temperature and fan shaft power consumption for a given set of ambient conditions and design parameters. The induced draught system generally requires more fan shaft power than the forced draught system for a given heat rejection rate as is seen from table F.1 in appendix F. This is so because the fan draws hot air, which has a low density, through the tube bundle.

After outlining the method for data presentation in chapter 3, a presentation of experimental and calculated results performed for the different finned tubes is given in chapter 4. It is concluded from figures A.5 through to A.7 in appendix A that there is an increase in the heat transfer parameter with a decrease in the fin pitch. The decrease in fin pitch however also results in an increase in the pressure drop across the bundle. Varying the water mass flowrate has a negligible effect on the heat transfer parameter, everything else being held constant. A less than 1 % variation (figure A.8) of the Ny-value, the heat transfer parameter, was obtained by performing such a test. Furthermore, thermal-flow characteristics of the different finned tubes (see figures 3.1 and 3.2) are presented in appendices C, D and E.

Finally, in chapter 5, different heat exchanger bundles are installed in a forced draught ACHE with the view to comparing their respective thermal-flow performance. Performance characteristics of a reference extruded finned round tube as tested by Kröger [98KR1] is also presented. Comparison is also made between the experimentally

determined heat transfer characteristics for the R-2.3D finned tube tested and the theoretical prediction by Ganguli [85GA1]. It is shown in figure 5.1 that the R-2.3D finned tube, due to its poor quality (as seen from the high contact resistance calculated), has a lower heat transfer parameter when compared to that of the reference extruded finned round tube, which compares very well with Ganguli correlation. It is also shown that the thermal contact resistance of the R-2.3D finned tube, at $R_c = 1.7219 \times 10^{-4}$ $m^2K/W$, is almost five times higher than the specified value for the reference round tube which has a value of $R_c = 4 \times 10^{-5}$ $m^2K/W$. The McQuiston correlation was also used to determine the heat transfer and airside pressure drop characteristics for a staggered plate-finned round tubes heat exchanger. The heat exchanger bundle was then incorporated in a forced draught ACHE and a thermal-flow performance thereof evaluated.

Lastly, using the computer performance evaluation program for a forced draught ACHE, it is shown in table 5.2 that the AE-2.5 DE finned tube heat exchanger requires the least amount of fan shaft power of $P_F = 17.9184$ kW for a specified heat rejection rate of $Q_{rem} = 5.022$ MW at a water mass flowrate of 100 kg/s. Less process fluid pumping power is required as it has the smallest pressure drop across the tube of $\Delta p_f = 3617.18$ Pa. The two ED-type finned tube heat exchanger also have better performance characteristics when compared to the standard reference extruded finned round tube heat exchanger bundle. The R-2.3D and the F-3.0 finned tube bundles perform poorly when compared to the standard reference extruded finned round tube bundles.

Based on the above conclusions, it is recommended that the use of elliptical finned tubes in a petrochemical industry as a better alternative, performance-wise, to the conventional finned round tube heat exchanger be considered for application where the process fluid pressure is not excessive.

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# APPENDIX A

## Effect of fin pitch and water mass flowrate on the thermal-flow performance of a heat exchanger bundle

## Table A.1.1: Tube specifications.

**Name:** AE-2.2 (2.2 mm fin pitch, 0.48875 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 4.075 | 214 | 468.5 | 2.18925 |
| 2 | 4.055 | 213 | 468 | 2.19718 |
| 3 | 4.01 | 209 | 469 | 2.24402 |
| 4 | 4.06 | 214 | 466 | 2.17757 |
| 5 | 4.05 | 214 | 467 | 2.18224 |
| 6 | 4.05 | 211 | 468 | 2.21801 |
| 7 | 4.005 | 209 | 461 | 2.20574 |
| 8 | 4.055 | 214 | 467 | 2.18224 |
| 9 | 4.045 | 209 | 468 | 2.23923 |
| 10 | 4.075 | 214 | 469 | 2.19159 |
| Average | 4.048 | 212.1 | 467.15 | 2.2027085 |

## Table A.1.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 24.95925 | 37.33766 | 65.5401 | 64.03753 | 100103.6 | 2.4724873 | 4.28044 |
| 2 | 25.28283 | 39.08679 | 65.33853 | 63.99107 | 100098.33 | 1.9706344 | 4.2590853 |
| 3 | 25.21611 | 41.1575 | 65.36698 | 64.17437 | 100099.42 | 1.5280545 | 4.260266 |
| 4 | 25.25604 | 43.83281 | 65.37377 | 64.35564 | 100098.77 | 1.1382156 | 4.2780567 |
| 5 | 25.52157 | 46.3568 | 65.50876 | 64.6232 | 100094.44 | 0.8901218 | 4.2864099 |
| 6 | 25.47576 | 49.96894 | 65.70552 | 64.97584 | 100095.19 | 0.638672 | 4.279248 |

## Table A.1.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 462 | 431 | 1298 |
| 2 | 323 | 740 | 865 |
| 3 | 213 | 445 | 540 |
| 4 | 134 | 248 | 316 |
| 5 | 92 | 153 | 205 |
| 6 | 57 | 80 | 115 |

## Table A.1.3: Pressure drops for isothermal tests.

($P_{atm}$ = 100820 N/m$^2$, $T_{aiwb}$ = 17.5°C, $T_{aidb}$ = 23.5°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 426 | 400 | 1200 |
| 2 | 320 | 756 | 890 |
| 3 | 210 | 454 | 543 |
| 4 | 131 | 250 | 312 |
| 5 | 89 | 154 | 215 |
| 6 | 56 | 82 | 110 |

**Table A.1.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 33.35312136 | 0.9810046 | 31364.529 | 29521.918 | 0.9412517 | 5.8748272 |
| 2 | 32.08727217 | 0.9806031 | 27875.744 | 26607.73 | 0.9545119 | 4.5488069 |
| 3 | 31.0128674 | 0.9807224 | 24964.196 | 23853.161 | 0.9554949 | 4.4505141 |
| 4 | 29.46764403 | 0.9816576 | 21670.896 | 20827.487 | 0.961081 | 3.8918952 |
| 5 | 27.96770359 | 0.9828238 | 19007.581 | 18489.283 | 0.972732 | 2.7267952 |
| 6 | 25.84409028 | 0.9845631 | 16034.619 | 15665.804 | 0.9769989 | 2.3001126 |

**Table A.1.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 193444.9179 | 17554.869 | 570208.1 | 183295.5 | 9.4499855 |
| 2 | 192132.3063 | 17469.291 | 453279.85 | 168192.98 | 10.365316 |
| 3 | 192482.3403 | 17483.451 | 350622.99 | 154756.02 | 11.331016 |
| 4 | 193550.921 | 17553.879 | 260305.97 | 140070.54 | 12.790973 |
| 5 | 194497.1067 | 17601.939 | 202867.18 | 128323.16 | 14.294931 |
| 6 | 194947.217 | 17603.279 | 144935.72 | 115935.98 | 17.104573 |

**Correlating equations:** $Ny = 2164.3479Ry^{0.334505}$

$K_{he} = 2621.945Ry^{-0.425420}$

**Table A.1.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 572576.0191 | 9.2199913 |
| 2 | 479721.4624 | 9.8664041 |
| 3 | 373010.7934 | 10.709365 |
| 4 | 277427.0139 | 12.077054 |
| 5 | 217746.82 | 13.31906 |
| 6 | 158626.428 | 15.791528 |

**Correlating equation:** $K_{heiso} = 2099.6263Ry^{-0.410382}$

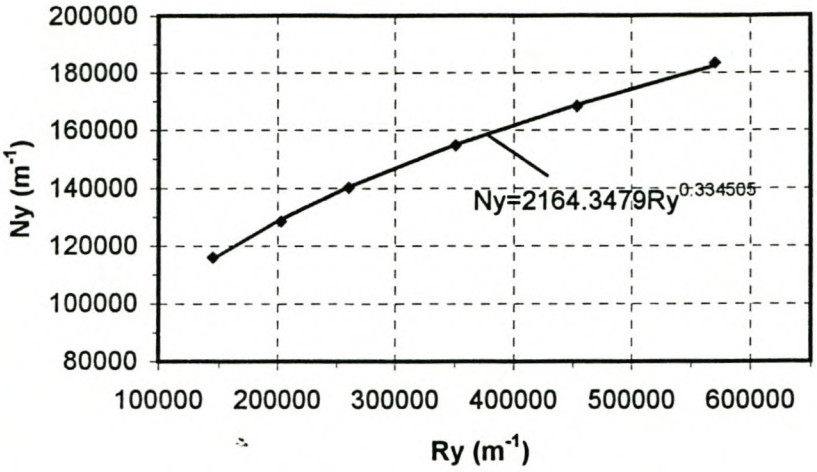



**Figure A.1.1: Heat transfer parameter.**

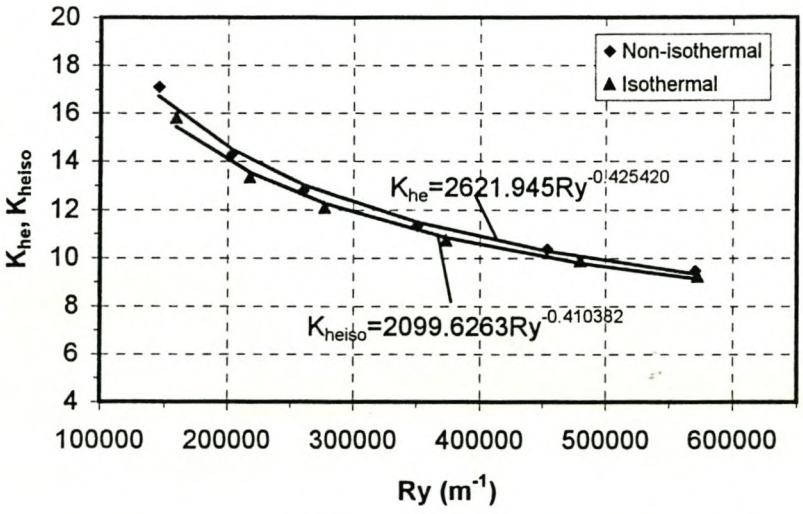



**Figure A.1.2: Pressure drop coefficient.**

## Table A.2.1: Tube specifications.

**Name:** AE-2.3 (2.3 mm fin pitch, 0.463920 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 3.885 | 201 | 468 | 2.32836 |
| 2 | 3.91 | 201 | 468 | 2.32836 |
| 3 | 3.92 | 202 | 466 | 2.30693 |
| 4 | 3.665 | 201 | 468 | 2.32836 |
| 5 | 3.875 | 200 | 465 | 2.32500 |
| 6 | 3.9 | 203 | 468 | 2.30542 |
| 7 | 3.885 | 200 | 468 | 2.34000 |
| 8 | 3.89 | 199 | 468 | 2.35176 |
| 9 | 3.885 | 201 | 467 | 2.32338 |
| 10 | 3.88 | 200 | 470 | 2.35000 |
| Average | 3.8695 | 200.8 | 467.6 | 2.3287566 |

## Table A.2.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 26.94005 | 37.55132 | 63.06423 | 61.73734 | 100101.21 | 2.4738871 | 4.3597672 |
| 2 | 27.4969 | 39.13672 | 62.97453 | 61.76826 | 100092.14 | 1.9831436 | 4.3347139 |
| 3 | 27.84769 | 41.08774 | 62.95214 | 61.90222 | 100086.43 | 1.5330234 | 4.3192619 |
| 4 | 27.84497 | 43.23277 | 62.86764 | 61.96936 | 100086.48 | 1.1482015 | 4.3274 |
| 5 | 27.94133 | 45.49372 | 62.85996 | 62.08626 | 100084.91 | 0.8674404 | 4.2976692 |
| 6 | 27.97915 | 48.54143 | 63.11699 | 62.48605 | 100084.29 | 0.6318639 | 4.3457313 |

## Table A.2.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 432 | 432 | 1294 |
| 2 | 302 | 750 | 864 |
| 3 | 197.5 | 448 | 534 |
| 4 | 124.5 | 252 | 313 |
| 5 | 82 | 145 | 190 |
| 6 | 50.5 | 78 | 110 |

## Table A.2.3: Pressure drops for isothermal tests.

($P_{atm}$ = 100360 N/m$^2$, $T_{aiwb}$ = 26°C, $T_{aidb}$ = 27°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 439 | 438 | 1335 |
| 2 | 295 | 742 | 853 |
| 3 | 198.5 | 462 | 546 |
| 4 | 123.5 | 258 | 316 |
| 5 | 83.5 | 158 | 202 |
| 6 | 53.5 | 88 | 118 |

**Table A.2.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 29.92034008 | 0.9824207 | 26923.339 | 25970.645 | 0.9646146 | 3.5385442 |
| 2 | 28.74546391 | 0.9817977 | 23671.474 | 23632.78 | 0.9983654 | 0.1634642 |
| 3 | 27.51795631 | 0.9818705 | 20813.273 | 20722.065 | 0.9956178 | 0.4382224 |
| 4 | 26.22460502 | 0.9825785 | 18118.657 | 18014.38 | 0.9942448 | 0.575521 |
| 5 | 24.82845077 | 0.9836653 | 15614.395 | 15649.709 | 1.0022616 | -0.226158 |
| 6 | 23.14152432 | 0.9855811 | 13325.452 | 13228.622 | 0.9927334 | 0.7266604 |

**Table A.2.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 190214.7618 | 17593.975 | 569270.13 | 174143.61 | 8.7900039 |
| 2 | 189038.7672 | 17505.791 | 455089.91 | 158271.9 | 9.5293375 |
| 3 | 188521.755 | 17458.681 | 350781.92 | 144297.7 | 10.390061 |
| 4 | 188852.5011 | 17485.548 | 262045.44 | 130773.86 | 11.63515 |
| 5 | 187707.8354 | 17389.598 | 197401.12 | 118080.69 | 13.375857 |
| 6 | 190737.1445 | 17585.896 | 143256.93 | 107045.17 | 15.448128 |

**Correlating equations:** $Ny = 1646.0652Ry^{0.350891}$

$K_{he} = 1948.849Ry^{-0.408734}$

**Table A.2.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 590141.2328 | 8.6929661 |
| 2 | 468649.6864 | 9.262767 |
| 3 | 370937.0728 | 9.948908 |
| 4 | 277829.1369 | 11.033842 |
| 5 | 217447.0658 | 12.178527 |
| 6 | 162025.0696 | 14.054155 |

**Correlating equation:** $K_{heiso} = 1143.0382Ry^{-0.368724}$

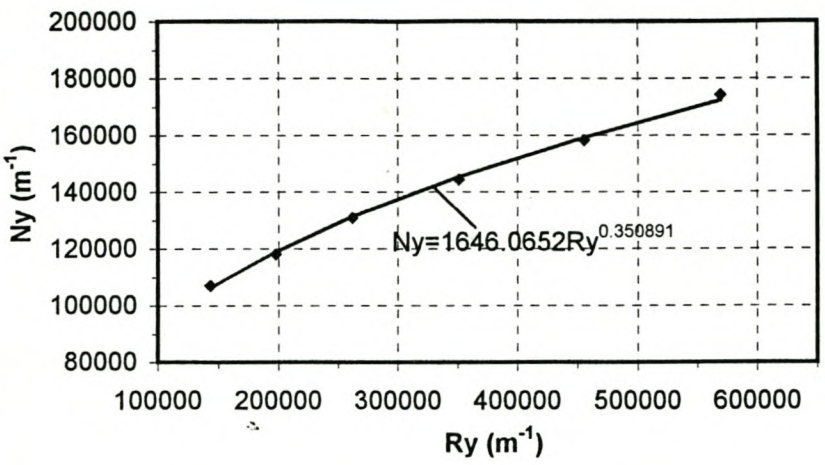



**Figure A.2.1: Heat transfer parameter.**

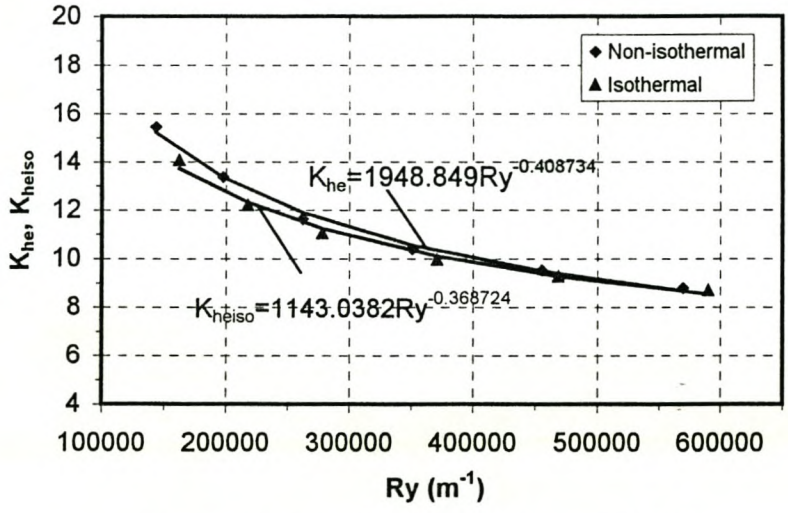



**Figure A.2.3: Pressure drop coefficient.**

**Table A.3.1: Tube specifications.**

**Name:** AE-2.4 (2.4 mm fin pitch, 0.50417 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 3.845 | 193 | 468 | 2.425 |
| 2 | 3.845 | 192 | 468 | 2.438 |
| 3 | 3.785 | 193 | 470 | 2 435 |
| 4 | 3.865 | 193 | 468 | 2.425 |
| 5 | 3.835 | 192 | 470 | 2.448 |
| 6 | 3.87 | 193 | 470 | 2.435 |
| 7 | 3.835 | 192 | 468 | 2.438 |
| 8 | 3.86 | 192 | 468 | 2.438 |
| 9 | 3.835 | 193 | 470 | 2.435 |
| 10 | 3.895 | 193 | 469 | 2.430 |
| Average | 3.847 | 192.6 | 468.9 | 2.4345909 |

**Table A.3.2a: Test data.**

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 28.35868 | 37.0178 | 60.94835 | 58.32739 | 100088.06 | 2.4879917 | 2.009746111 |
| 2 | 29.02352 | 38.57636 | 61.03271 | 58.67113 | 100077.23 | 1.9857809 | 2.003994642 |
| 3 | 29.47976 | 40.48772 | 61.16273 | 59.0604 | 100069.8 | 1.5432355 | 2.011583053 |
| 4 | 29.51082 | 42.52892 | 61.2469 | 59.39182 | 100069.3 | 1.1484482 | 1.998794526 |
| 5 | 29.76281 | 44.84549 | 61.493 | 59.90401 | 100065.19 | 0.858496 | 2.003603695 |
| 6 | 29.84254 | 46.68411 | 61.4953 | 60.10833 | 100063.89 | 0.676669 | 1.988474838 |

**Table A.3.2b: Test data.**

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 411 | 437 | 1286 |
| 2 | 286.5 | 752 | 854 |
| 3 | 190 | 454 | 534 |
| 4 | 118 | 252 | 308 |
| 5 | 75 | 142 | 182 |
| 6 | 53 | 89 | 120 |

**Table A.3.3: Pressure drops for isothermal tests.**

($P_{atm}$ = 100450 N/m$^2$, $T_{aiwb}$ = 25°C, $T_{aidb}$ = 25°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 416.5 | 446 | 1320 |
| 2 | 278 | 738 | 832 |
| 3 | 189 | 458 | 533 |
| 4 | 119 | 256 | 310 |
| 5 | 78.5 | 148 | 187 |

**Table A.3.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 26.83680248 | 0.9584434 | 22151.902 | 22103.588 | 0.9978189 | 0.218106 |
| 2 | 25.88607172 | 0.9601778 | 19502.112 | 19866.228 | 1.0186706 | -1.867059 |
| 3 | 24.86302887 | 0.9613991 | 17463.068 | 17760.2 | 1.0170148 | -1.701483 |
| 4 | 23.86656541 | 0.9631467 | 15369.696 | 15579.971 | 1.0136812 | -1.368117 |
| 5 | 22.73149989 | 0.9665464 | 13311.345 | 13387.441 | 1.0057166 | -0.571656 |
| 6 | 21.62681115 | 0.9696059 | 11716.025 | 11605.561 | 0.9905716 | 0.942841 |

**Table A.3.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 84093.32779 | 9025.1597 | 572861.24 | 171780.43 | 8.2406326 |
| 2 | 84128.36029 | 9015.5347 | 455914.62 | 154523.78 | 8.9851654 |
| 3 | 84782.99946 | 9058.7982 | 353252.93 | 142369.65 | 9.8287332 |
| 4 | 84511.55686 | 9022.0805 | 262222.48 | 128973.72 | 10.98527 |
| 5 | 85204.96741 | 9061.4643 | 195399.4 | 115514.51 | 12.443564 |
| 6 | 84694.28324 | 9009.7728 | 153656.25 | 105677.36 | 14.110631 |

**Correlating equations:** $Ny = 1426.4684Ry^{0.360576}$

$K_{he} = 1664.625Ry^{-0.4011}$

**Table A.3.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 600748.8212 | 8.0990687 |
| 2 | 471547.1426 | 8.7740578 |
| 3 | 372600.158 | 9.5539214 |
| 4 | 279189.3667 | 10.714086 |
| 5 | 212291.8196 | 12.22388 |

**Correlating equation:** $K_{heiso} = 1510.285Ry^{-0.393805}$

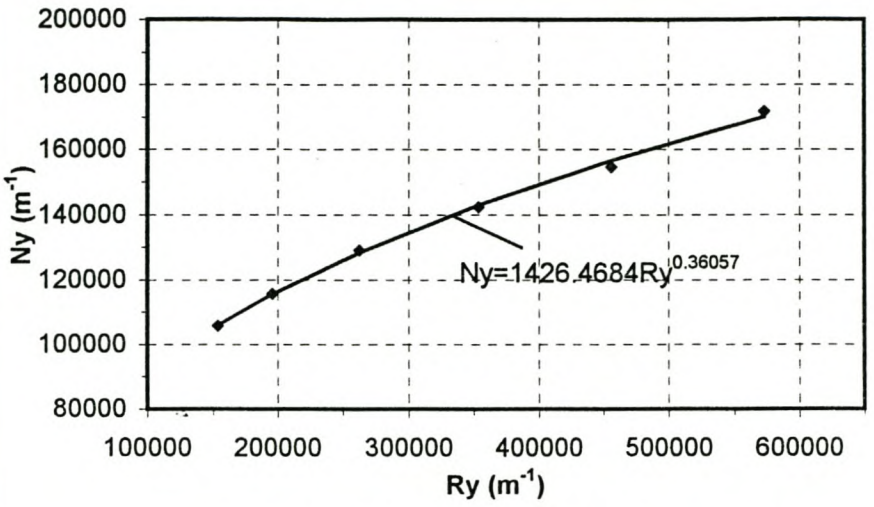



**Figure A.3.1: Heat transfer parameter.**

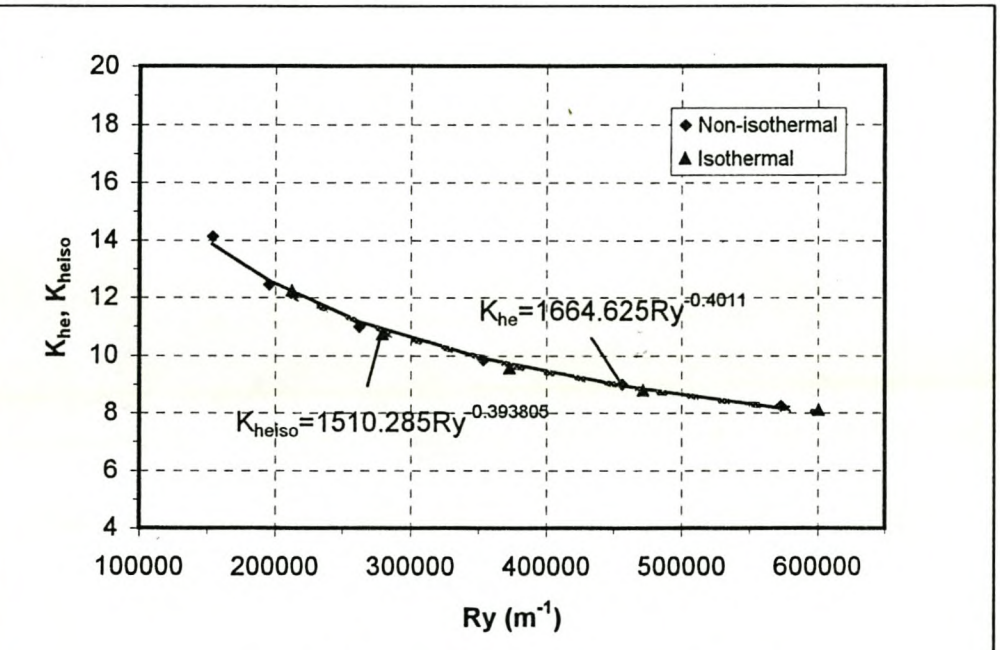



**Figure A.3.2: Pressure drop coefficient.**

**Table A.4.1: Tube specifications.**

**Name:** AE-2.5 (2.5 mm fin pitch, 0.5252 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 3.785 | 187 | 469 | 2.5080 |
| 2 | 3.79 | 187 | 467 | 2.4973 |
| 3 | 3.815 | 187 | 467 | 2.4973 |
| 4 | 3.805 | 187 | 466 | 2.4920 |
| 5 | 3.72 | 187 | 470 | 2.5134 |
| 6 | 3.73 | 188 | 469 | 2.4947 |
| 7 | 3.715 | 187 | 467 | 2.4973 |
| 8 | 3.71 | 187 | 467 | 2.4973 |
| 9 | 3.705 | 187 | 467 | 2.4973 |
| 10 | 3.72 | 187 | 469 | 2.5080 |
| Average | 3.7495 | 187.1 | 467.8 | 2.5002702 |

**Table A.4.2a: Test data.**

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 28.61717 | 37.51778 | 62.65179 | 59.89692 | 99844.965 | 2.4999128 | 1.971599974 |
| 2 | 28.60199 | 38.77568 | 62.45247 | 60.02759 | 99845.211 | 1.9946626 | 1.991749573 |
| 3 | 28.61008 | 40.38286 | 62.63118 | 60.45633 | 99845.08 | 1.5332256 | 1.982006204 |
| 4 | 28.43224 | 42.3869 | 62.74792 | 60.81269 | 99847.97 | 1.1534004 | 1.982771951 |
| 5 | 28.30862 | 44.36552 | 62.67126 | 60.9669 | 99849.979 | 0.8879161 | 1.956566806 |
| 6 | 27.90576 | 46.8413 | 62.82766 | 61.33433 | 99856.525 | 0.6504746 | 1.965057332 |

**Table A.4.2b: Test data.**

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 398.5 | 440 | 1275 |
| 2 | 277 | 756 | 844 |
| 3 | 181 | 446 | 518 |
| 4 | 114 | 253 | 304 |
| 5 | 76 | 151 | 190 |
| 6 | 48 | 82 | 110 |

**Table A.4.3: Pressure drops for isothermal tests.**

($P_{atm}$ = 100010 N/m$^2$, $T_{aiwb}$ = 22.5°C, $T_{aidb}$ = 28°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 401 | 449 | 1302 |
| 2 | 275 | 764 | 848 |
| 3 | 179 | 453 | 520 |
| 4 | 112 | 254 | 302 |
| 5 | 75 | 155 | 191 |
| 6 | 45 | 80 | 104 |

**Table A.4.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 28.09433824 | 0.9589205 | 22659.027 | 22595.246 | 0.9971852 | 0.2814792 |
| 2 | 27.36783627 | 0.9619296 | 20666.23 | 20075.477 | 0.9714146 | 2.858539 |
| 3 | 26.76005056 | 0.9636161 | 18382.988 | 17904.27 | 0.9739587 | 2.6041331 |
| 4 | 25.90642774 | 0.9652182 | 16393.787 | 15923.402 | 0.9713071 | 2.8692855 |
| 5 | 24.79186633 | 0.9676208 | 14522.956 | 13822.497 | 0.9517688 | 4.8231162 |
| 6 | 23.64276977 | 0.9706595 | 12549.22 | 12147.414 | 0.9679816 | 3.2018416 |

**Table A.4.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 84578.12287 | 8971.5453 | 571375.79 | 166964.42 | 7.9382045 |
| 2 | 85398.23932 | 9046.4515 | 455207.66 | 154331.36 | 8.6497517 |
| 3 | 85370.50143 | 9026.0089 | 349214.03 | 138611.3 | 9.5408682 |
| 4 | 85707.75249 | 9041.8831 | 262132.79 | 126233.53 | 10.586997 |
| 5 | 84624.25133 | 8943.7275 | 201349.83 | 115646.5 | 11.873808 |
| 6 | 85325.82056 | 8990.4399 | 147151.61 | 103375.31 | 13.926268 |

**Correlating equations:** $Ny = 1556.625Ry^{0.352482}$

$K_{he} = 1168.125Ry^{-0.37809}$

**Table A.4.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 593021.2925 | 7.7447379 |
| 2 | 471883.865 | 8.388134 |
| 3 | 364581.4855 | 9.1467484 |
| 4 | 273596.2732 | 10.162501 |
| 5 | 213729.9005 | 11.151509 |
| 6 | 153253.4224 | 13.013529 |

**Correlating equation:** $K_{heiso} = 1168.125Ry^{-0.37809}$

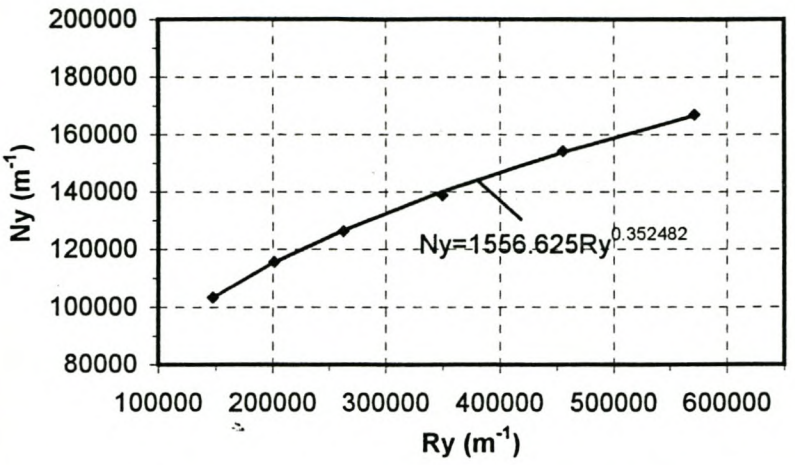



**Figure A.4.1: Heat transfer parameter.**

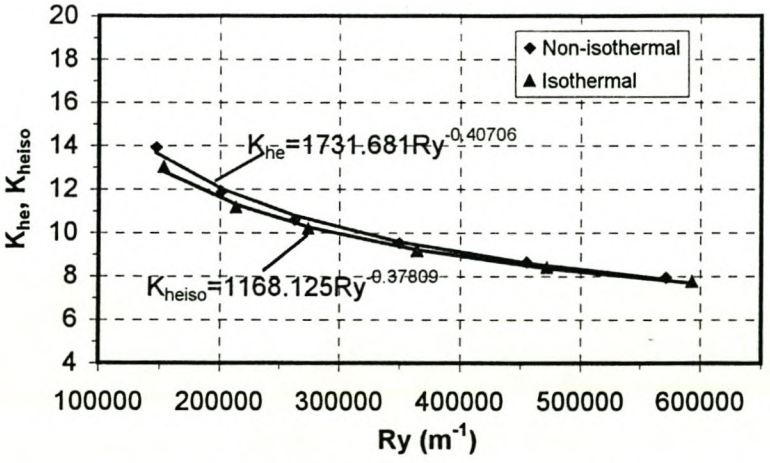



**Figure A.4.2: Pressure drop coefficient.**

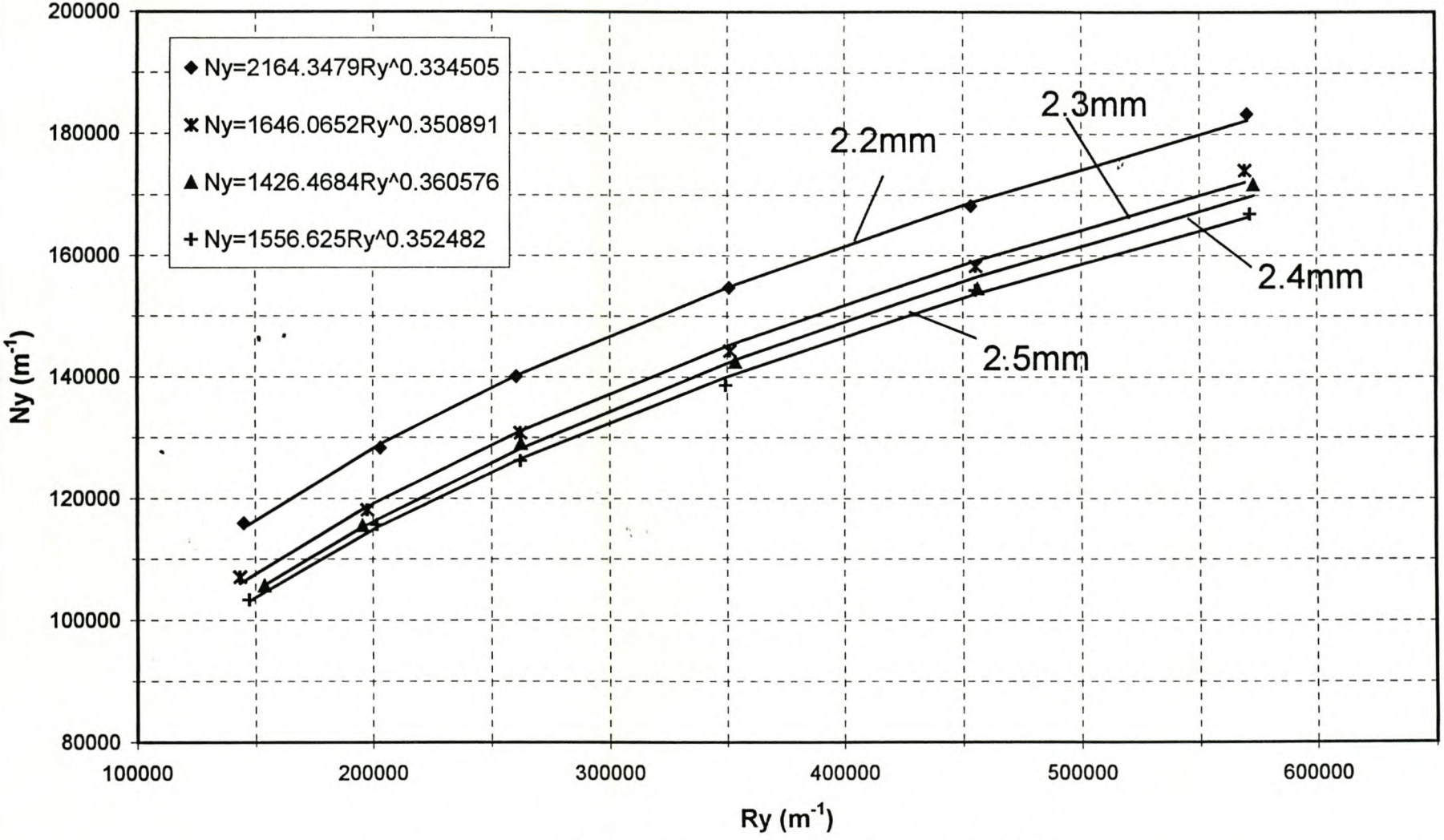



**Figure A.5: Heat transfer parameter.**

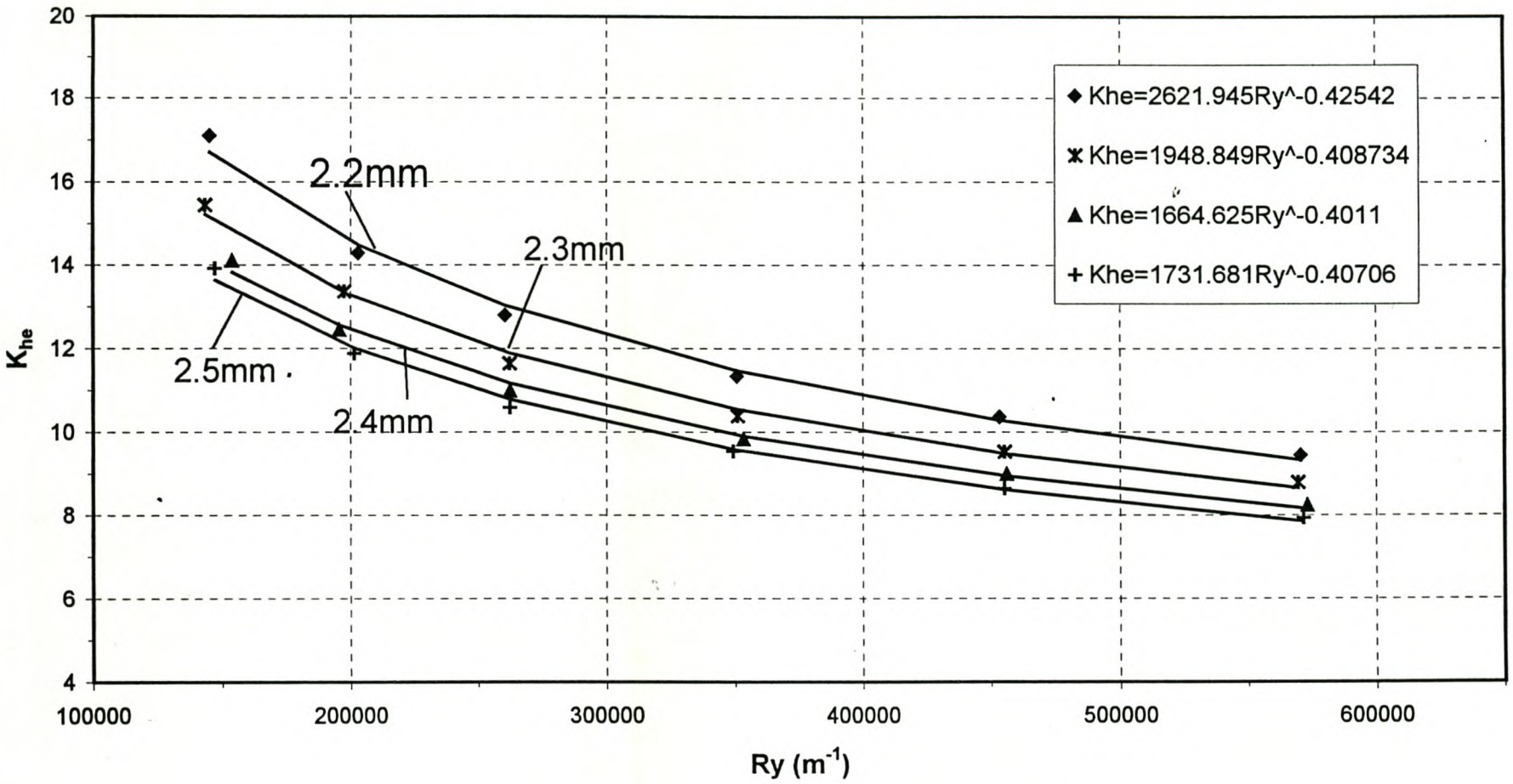



**Figure A.6: Non-isothermal pressure drop.**

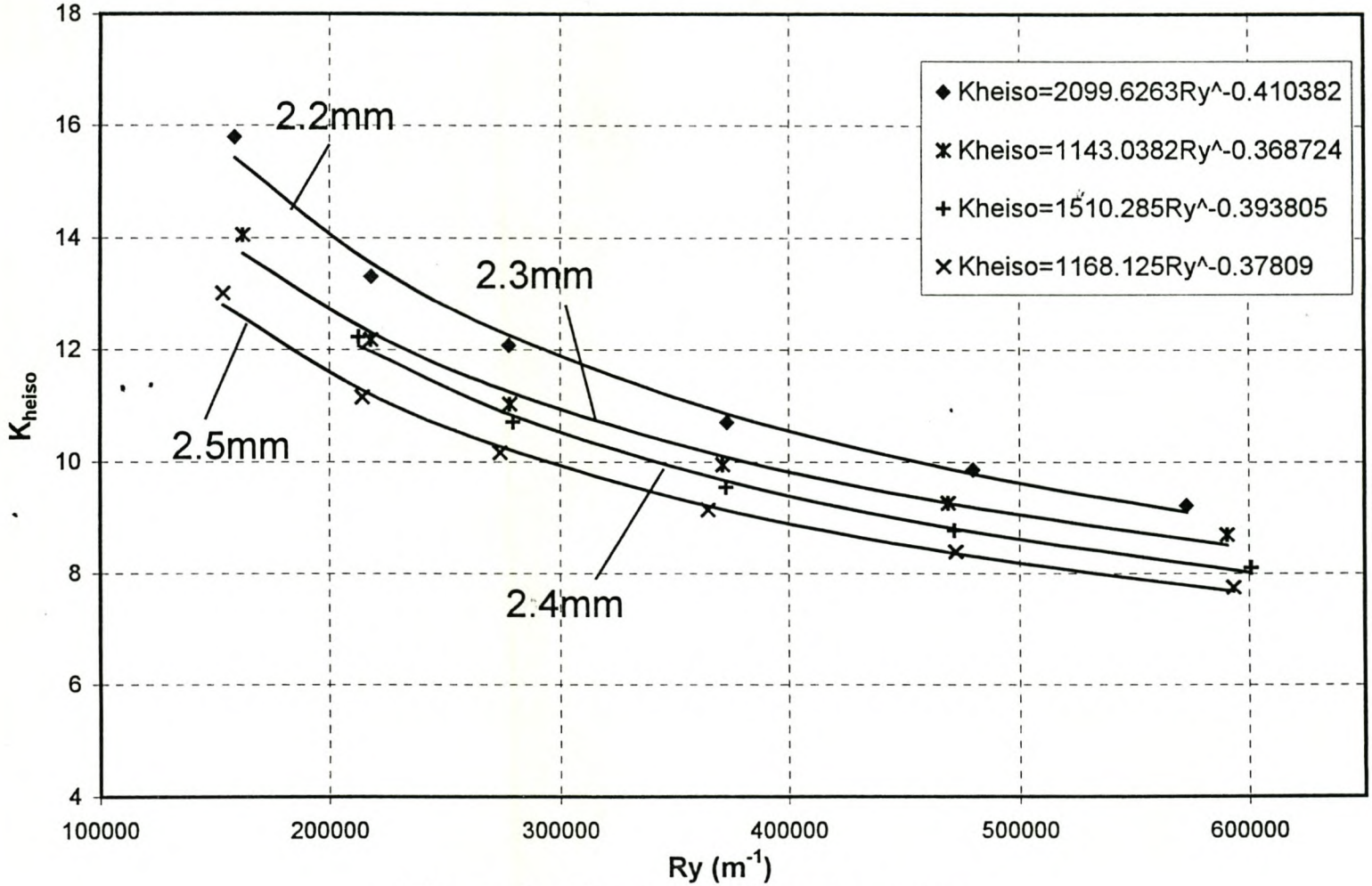



**Figure A.7: Isothermal pressure drop.**

## Testing heat exchanger bundle at different water flow rates.

### Table A.5.1a: Test data.

Name: AE-2.4 (2.4 mm fin pitch, 0.50417 mm fin thickness)

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 27.11008 | 41.24413 | 60.69384 | 59.38481 | 100048.66 | 1.1558117 | 3.1432124 |
| 2 | 27.39349 | 41.30084 | 60.76997 | 59.19934 | 100044.05 | 1.1557197 | 2.5665074 |
| 3 | 27.94464 | 41.42713 | 60.62866 | 58.80418 | 100035.08 | 1.155517 | 2.1108332 |
| 4 | 28.67481 | 41.75583 | 60.76402 | 58.63624 | 100023.19 | 1.1572083 | 1.7496198 |
| 5 | 29.24338 | 41.97449 | 61.02282 | 58.51398 | 100013.94 | 1.1522899 | 1.4503038 |

### Table A.5.1b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 118 | 254 | 310 |
| 2 | 118 | 254 | 310 |
| 3 | 118 | 254 | 309 |
| 4 | 118 | 255 | 310 |
| 5 | 117 | 253 | 309 |

### Table A.5.2a: Test results.

| Run no. | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 25.321537 | 0.9949909 | 16768.073 | 16952.775 | 1.0110151 | -1.101507 |
| 2 | 25.133255 | 0.9938396 | 16496.348 | 16651.412 | 1.0093999 | -0.939991 |
| 3 | 24.56972 | 0.9925507 | 15986.84 | 15936.335 | 0.9968408 | 0.3159188 |
| 4 | 24.069355 | 0.9909735 | 15530.04 | 15429.447 | 0.9935226 | 0.6477355 |
| 5 | 23.793178 | 0.9891201 | 15047.752 | 15102.257 | 1.0036222 | -0.362219 |

### Table A.5.2b: Test results.

| Run no. | $Re_w$ --- | $h_w$ W/m$^2$ K | Ry m$^{-1}$ | Ny m$^{-1}$ |
|---|---|---|---|---|
| 1 | 132332.06 | 13173.928 | 264833.6 | 125666.84 |
| 2 | 107961.97 | 11105.96 | 264685.28 | 125872.01 |
| 3 | 88429.62 | 9409.2413 | 264386.72 | 126246.72 |
| 4 | 73278.91 | 8037.542 | 264388.16 | 126805.98 |
| 5 | 60806.358 | 6870.4 | 262978.24 | 126108.99 |

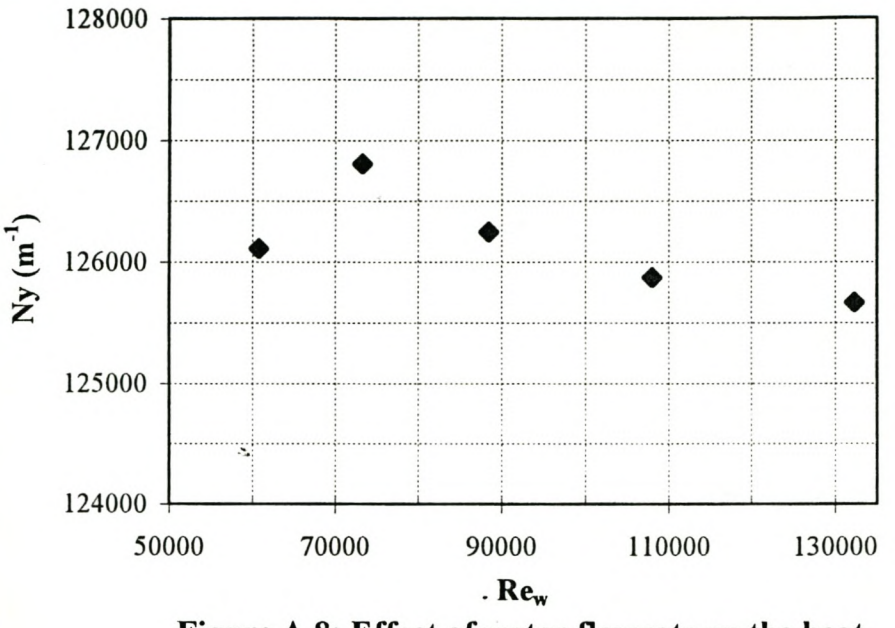



**Figure A.8: Effect of water flowrate on the heat transfer parameter.**

From the above figure, it is noticed that there is less than 1% variation on the Ny-value when testing at different water flow rates.

# APPENDIX B

## Sample calculation for determining the heat transfer characteristics for a single pass air-cooled heat exchanger bundle

# Sample calculation for determining the heat transfer characteristics of a heat exchanger bundle.

**Finned tube bundle specification (*2.4mm fin pitch*).**

The following sample calculation shows how the heat transfer characteristics for the elliptical tube air-cooled heat exchanger bundle, consisting of one tube-pass, single tube row, is determined.

Test Data: Run no.3 in Appendix A (Table A.5)

The finned tube bundle specifications used for this test are:

| | |
|---|---|
| Bundle width: $W_b$ | 0.47 m |
| Bundle height: $H_b$ | 0.50 m |
| Number of tubes per row: $n_{tr}$ | 10.0 |
| Number of rows: $n_r$ | 1.00 |
| Frontal area: $A_{fr}$ | 0.235 $m^2$ |
| Tube length per pass: $L_t$ | 0.47 m |
| Tube length along all passes: L | 4.70 m |

*(for waterside heat transfer coefficient)*

| | |
|---|---|
| Hydraulic diameter: $d_e$ | 0.0259221 m |
| Waterside surface area per tube: $A_w$ | 0.096464 $m^2$ |

*(for determination of air mass flowrate)*

| | |
|---|---|
| Nozzle diameter: $d_n$ | 0.2509 m |
| Windtunnel upstream cross-section: $A_{tus}$ | 1.44 $m^2$ |

Table B.1 below shows temperature as well as pressure readings used for this sample calculation.

Table B.1: Test data.

| $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C |
|---|---|---|---|
| 27.94464 | 41.42713 | 60.62866 | 58.80418 |
| $p_{meas}$ Pa | $\Delta p_{bundle}$ Pa | $\Delta p_n$ Pa | $\Delta p_{up}$ Pa |
| 100490 | 118 | 254 | 309 |

The wetbulb temperature of the inlet air is, $T_{aiwb} = 26.5\ ^{\circ}C$

A corrected ambient pressure value is given by the following equation from the barometer used.

$p_{amb} = p_{meas} - 0.000162 \times p_{meas} \times T_{ai} = 100035.154\ N/m^2$

Hence pressure upstream of the nozzle is

$p_u = p_{amb} - \Delta p_{up} = 100035.154 - 309 = 99726.154\ N/m^2$

The following thermophysical properties of the airstream are required to determine air mass flowrate through the flow nozzles in the windtunnel.

Water vapour pressure at 26.5 °C (299.65 °K) from Kröger [98KR1],

$$Z = 10.79586\left(1 - \tfrac{273.16}{T}\right) + 5.02808\log_{10}\left(\tfrac{273.16}{T}\right) + 1.50474x10^{-4}\left[1 - 10^{-8.29692\left(\frac{T}{273.16}-1\right)}\right]$$

$$+ 4.2873x10^{-4}\left[10^{4.76955\left(1-\frac{273.16}{T}\right)} - 1\right] + 2.786118312$$

$$= 10.79586\left(1 - \tfrac{273.16}{299.65}\right) + 5.02808\log_{10}\left(\tfrac{273.16}{299.65}\right) + 1.50474x10^{-4}\left[1 - 10^{-8.29692\left(\frac{299.65}{273.16}-1\right)}\right]$$

$$+ 4.2873x10^{-4}\left[10^{4.76955\left(1-\frac{273.16}{299.65}\right)} - 1\right] + 2.786118312$$

$$= 3.53922$$

Therefore,

$$p_v = 10^{z} = 10^{3.53922} = 3461.16\ N/m^2$$

Humidity ratio is calculated using the following equation

$$w = \frac{\left[\{2501.6 - 2.3263(T_{wb} - 273.15)\} x \left\{\frac{0.62509 p_{vwb}}{p_{abs} - 1.005 p_{vwb}}\right\} - 1.00416(T - T_{wb})\right]}{\left[2501.6 + 1.8577(T - 273.15) - 4.184(T_{wb} - 273.15)\right]}$$

$$= \frac{\left[\{2501.6 - 2.3263(299.65 - 273.15)\}x\left\{\frac{0.62509x3461.16}{100035.154 - 1.005x3461.16}\right\} - 1.00416(301.09 - 299.65)\right]}{[2501.6 + 1.8577(301.09 - 273.15) - 4.184(299.65 - 273.15)]}$$

$= 0.0223824$ kg/kg dry air

Air density after bundle and before nozzle at 314.577 °K is

$$\rho_{avo} = (1+w)\left\{1 - \frac{w}{w + 0.62198}\right\}x\frac{p_u}{287.08T}$$

$$= (1 + 0.0223824)\left\{1 - \frac{0.0223824}{0.0223824 + 0.62198}\right\}x\frac{99726.154}{287.08x314.577}$$

$$= 1.089779 \text{ kg/m}^3$$

Dynamic viscosity of dry air is

$$\mu_{ao} = 2.287973x10^{-6} + 6.259793x10^{-8}T - 3.131956x10^{-11}T^2 + 8.15038x10^{-15}T^3$$

$$= 2.287973x10^{-6} + 6.259793x10^{-8}(314.577) - 3.131956x10^{-11}(314.577)^2 + 8.15038x10^{-15}(314.577)^3 = 1.9134x10^{-5} \text{ kg/ms}$$

Dynamic viscosity of water vapour in air,

$$\mu_{vo} = 2.562435x10^{-6} + 1.816683x10^{-8}T + 2.579066x10^{-11}T^2 - 1.067299x10^{-14}T^3$$

$$= 2.562435x10^{-6} + 1.816683x10^{-8}(314.577) + 2.579066x10^{-11}(314.577)^2 - 1.067299x10^{-14}(314.577)^3 = 1.04973x10^{-5} \text{ kg/ms}$$

Calculating the dynamic viscosity of the air-vapour mixtures,

$$X_a = \frac{1}{1 + 1.608w} = \frac{1}{1 + 1.608x0.0223824} = 0.965259$$

$$X_v = \frac{w}{w + 0.622} = \frac{0.0223824}{0.0223824 + 0.622} = 0.0347347$$

with these values, the dynamic viscosity of the air-vapour mixtures is

$$\mu_{avo} = \frac{X_a \mu_{ao} M_a^{0.5} + X_v \mu_{vo} M_v^{0.5}}{X_a M_a^{0.5} + X_v M_v^{0.5}}$$

$$= \frac{0.965259x1.9134x10^{-5} x28.97^{0.5} + 0.0347347x1.04973x10^{-5} x18.016^{0.5}}{0.965259x28.97^{0.5} + 0.0347347x18.016^{0.5}}$$

$$= 1.8895x10^{-5} \text{ kg/ms}$$

where $M_a = 28.97$ kg/mole, $M_v = 18.016$ kg/mole, $X_a = 1/(1+1.608w)$ and $X_v = w/(w+0.622)$

For this particular test, only one nozzle was open in the windtunnel. The air mass flow is given by equation (3.3.1)

$$m_a = C_n \phi_g Y \pi d_n^2 (2\rho_{avo} \Delta p_n)^{0.5} / 4$$

The gas expansion factor is given by equation (3.3.3)

$$\phi_g = 1 - \frac{3\Delta p_n}{(4p_{up}\frac{c_p}{c_v})}$$

$$= 1 - \frac{3x254}{(4x99726.154x1.4)} = 0.99864$$

For a compressible fluid, it can be shown that the approach velocity factor is approximately,

$$Y = 1 + 0.5(A_n / A_{tus})^2 + 2(A_n / A_{tus})^2 \Delta p_n / (p_{up}\frac{c_p}{c_v})$$

$$= 1 + 0.5(\frac{\pi x0.2509^2}{4x1.44})^2 + 2(\frac{\pi x0.2509^2}{4x1.44})^2 x\frac{254}{99726.154x1.4} = 1.000594$$

The nozzle coefficient of discharge is a function of the nozzle Reynolds number. Initially, assume $C_n = 1$.

Therefore, air mass flowrate

$$m_a = 1x0.99864x1.000594x\pi x0.2509^2(2x1.089779x254)^{0.5}/4 = 1.1624 \text{ kg/s}$$

and the nozzle Re number

$$\text{Re}_n = \frac{4m_a}{\pi d_n \mu_{avo}} = \frac{4x1.1624}{\pi x0.2509x1.8895x10^{-5}} = 312192.06$$

Since $100000 \leq \text{Re}_n < 350000$, use equation. (3.3.2b) to find a new $C_n$.

$$C_n = 0.9758 + 1.08x10^{-7}(312675.3) - 1.6x10^{-13}(312675.3)^2$$

$$= 0.993926$$

A calculated new air mass flowrate is:

$$m_a = 0.993926x1.1624 = 1.15534 \text{ kg/s}$$

As before, a new $\text{Re}_n$ is calculated. Using equation. (3.3.2), a new coefficient of discharge, $C_n$ is again calculated. This is an iterative procedure to find $C_n$. The ultimate converged value of $C_n = 0.993901$ and the corresponding air mass flow $m_a = 1.155326$ kg/s.

Evaluate the arithmetic mean temperature of the air,

$$T_{am} = (T_{ai} + T_{ao})/2 = (301.09 + 314.577)/2 = 307.8335 \text{ K}$$

The specific heat of dry air is found using

$$c_{pa} = 1045.356 - 3.161783x10^{-1}T + 7.083814x10^{-4}T^2 - 2.705209x10^{-7}T^3$$

$$= 1045.356 - 3.161783x10^{-1}(307.8335) + 7.083814x10^{-4}(307.8335)^2$$

$$- 2.705209x10^{-7}(307.8335)^3 = 1007.262 \text{ J/kgK}$$

Dynamic viscosity of dry air is

$$\mu_a = 2.287973x10^{-6} + 6.259793x10^{-8}T - 3.131956x10^{-11}T^2 + 8.15038x10^{-15}T^3$$

$$= 2.287973x10^{-6} + 6.259793x10^{-8}(307.8335) - 3.131956x10^{-11}(307.8335)^2$$

$$+ 8.15038x10^{-15}(307.8335)^3 = 1.88276x10^{-5}\ \text{kg/ms}$$

Thermal conductivity of dry air is similarly calculated using the following equation

$$k_a = -4.937787x10^{-4} + 1.018087x10^{-4}T - 4.627937x10^{-8}T^2 + 1.250603x10^{-11}T^3$$

$$= -4.937787x10^{-4} + 1.018087x10^{-4}(307.8335) - 4.627937x10^{-8}(307.8335)^2$$

$$+ 1.250603x10^{-11}(307.8335)^3 = 2.68257x10^{-2}\ \text{W/mK}$$

Now, evaluating properties of water vapour in air at 307.8335 °K.

Specific heat of water vapour is

$$c_{pv} = 1.3605x10^3 + 2.31334T - 2.46784x10^{-10}T^5 + 5.91332x10^{-13}T^6$$

$$= 1.3605x10^3 + 2.31334(307.8335) - 2.46784x10^{-10}(307.8335)^5 + 5.91332x10^{-13}$$

$$x\ (307.8335)^6 = 1893.63\ J/kgK$$

Dynamic viscosity of water vapour is

$$\mu_v = 2.562435x10^{-6} + 1.816683x10^{-8}T + 2.579066x10^{-11}T^2 - 1.067299x10^{-14}T^3$$

$$= 2.562435x10^{-6} + 1.816683x10^{-8}(307.8335) + 2.579066x10^{-11}(307.8335)^2$$

$$- 1.067299x10^{-14}(307.8335)^3 = 1.02874x10^{-5}\ \text{kg/ms}$$

Thermal conductivity of water vapour is calculated using

$$k_v = 1.3046x10^{-2} - 3.756191x10^{-5}T + 2.217964x10^{-7}T^2 - 1.111562x10^{-10}T^3$$

$$= 1.3046x10^{-2} - 3.756191x10^{-5}(307.8335) + 2.217964x10^{-7}(307.8335)^2$$

$$-1.111562x10^{-10}(307.8335)^3 = 1.9258x10^{-2}\ \text{W/mK}$$

Finally, properties of air containing water vapour at 307.8335 °K are:

Specific heat of air is

$$c_{pav} = \frac{\left(c_{pa} + wc_{pv}\right)}{\left(1+w\right)} = \frac{\left(1007.262 + 0.0223824x1893.63\right)}{\left(1+0.0223824\right)} = 1026.667\,\text{J/kgK}$$

Thermal conductivity of air is

$$k_{av} = \frac{\left(X_a k_a M_a^{0.33} + X_v k_v M_v^{0.33}\right)}{\left(X_a M_a^{0.33} + X_v M_v^{0.33}\right)}$$

$$= \frac{\left(0.965259x2.68257x10^{-2}x28.97^{0.33} + 0.0347347x1.9258x10^{-2}x18.016^{0.33}\right)}{\left(0.965259x28.97^{0.33} + 0.0347347x18.016^{0.33}\right)}$$

$$= 2.6599x10^{-2}\ \text{W/mK}$$

Dynamic viscosity of air is found using

$$\mu_{av} = \frac{\left(X_a \mu_a M_a^{0.5} + X_v \mu_v M_v^{0.5}\right)}{\left(X_a M_a^{0.5} + X_v M_v^{0.5}\right)}$$

$$= \frac{\left(0.965259x1.88276x10^{-5}x28.97^{0.5} + 0.0347347x1.02874x10^{-5}x18.016^{0.5}\right)}{\left(0.965259x28.97^{0.5} + 0.0347347x18.016^{0.5}\right)}$$

$$= 1.859x10^{-5}\ \text{J/kgK}$$

Prandtl number of the air is given by,

$$\text{Pr}_{av} = \frac{\mu_{av} c_{pav}}{k_{av}} = \frac{1.859x10^{-5}x1026.667}{2.6599x10^{-2}} = 0.71761$$

Properties of water stream evaluated at the arithmetic mean temperature,

$$T_{wm} = (T_{wi} + T_{wo})/2 = (333.77866 + 331.95418)/2 = 332.866\,\text{K}$$

Density of water is found using the following equation

$$\rho_w = (1.49343x10^{-3} - 3.7164x10^{-6}T + 7.09782x10^{-9}T^2 - 1.90321x10^{-20}T^6)^{-1}$$

$$= (1.49343x10^{-3} - 3.7164x10^{-6}(332.866) + 7.09782x10^{-9}(332.866)^2 - 1.90321x10^{-20}$$

$$\text{x } (332.866)^6)^{-1} = 983.366\ \text{kg/m}^3$$

Specific heat of water is

$$c_{pw} = 8.15599x10^3 - 2.80627x10T + 5.11283x10^{-2}T^2 - 2.17582x10^{-13}T^6$$

$$= 8.15599x10^3 - 2.80627x10(332.866) + 5.11283x10^{-2}(332.866)^2 - 2.17582x10^{-13}$$

$$\text{x } (332.866)^6 = 4183.91 J/kgK$$

Dynamic viscosity of water is

$$\mu_w = 2.414x10^{-5}x10^{247.8/(T-140)} = 2.414x10^{-5}x10^{247.8/(332.866-140)}$$

$$= 4.612x10^{-4}\,\text{kg/ms}$$

Thermal conductivity of water is found from

$$k_w = -6.14255x10^{-1} + 6.9962x10^{-3}T - 1.01075x10^{-5}T^2 + 4.74737x10^{-12}T^4$$

$$= -6.14255x10^{-1} + 6.9962x10^{-3}(332.866) - 1.01075x10^{-5}(332.866)^2$$

$$+ 4.74737\text{x}10^{-12}(332.866)^4 = 0.65292\ \text{W/mK}$$

The Prandtl number of water is then found to be

$$\text{Pr}_w = \frac{\mu_w c_{pw}}{k_w} = \frac{4.612x10^{-4}x4183.91}{0.65292} = 2.9805$$

The heat transferred to the air stream is given by equation (3.4.9)

$$Q_a = m_a c_{pav}(T_{ao} - T_{ai}) = 1.155326x1026.667x(314.577 - 301.09) = 15997.404\,\text{W}$$

Similarly, the heat removed from the water stream is given by

$$Q_w = m_w c_{p_w}(T_{wi} - T_{wo}) = 2.1108x4183.91x(333.77866 - 331.95418) = 16112.71\,\text{W}$$

The two heat transfer values compare very well, i.e.

$$Q_a / Q_w = \frac{15997.404}{16112.71} = 0.99284$$

We now determine the waterside heat transfer coefficient. Firstly, calculating the waterside Reynolds number.

$$\text{Re}_w = \frac{m_w d_e}{A_{ts}\mu_w} = \frac{2.1108x0.0259221}{1330.082x10^{-6}x4.6512x10^{-4}} = 88445.112$$

From equation (3.4.15), it follows for smooth tubes that the friction factor is

$$f_w = (1.82\log_{10}\text{Re}_w - 1.64)^{-2} = (1.82\log_{10}(88445.112) - 1.64)^{-2} = 0.018446$$

From the Gnielinski equation (3.4.14), with $L_t = 0.47$m, the water-side heat transfer coefficient is,

$$h_w = \frac{k_w f_w(\text{Re}_w - 1000)x\,\text{Pr}_w\left[1 + (\frac{d_e}{L_t})^{0.67}\right]}{d_e x8\left[1 + 12.7\left(\frac{f_w}{8}\right)^{0.5}(\text{Pr}_w{}^{0.67} - 1)\right]}$$

$$= \frac{0.65292x0.018446(88445.112 - 1000)x2.9805\left[1 + (\frac{0.0259221}{0.47})^{0.67}\right]}{0.0259221x8\left[1 + 12.7\left(\frac{0.018446}{8}\right)^{0.5}(2.9805^{0.67} - 1)\right]}$$

$$= 10404.68\ \text{W/m}^2\text{K}$$

Furthermore, the logarithmic mean temperature difference (LMTD) is given by equation (3.4.8)

$$\Delta T_{lm} = \frac{(T_{wo} - T_{ai}) - (T_{wi} - T_{ao})}{\ln\left[\frac{(T_{wo} - T_{ai})}{(T_{wi} - T_{ao})}\right]}$$

$$= \frac{(331.95418 - 301.09) - (333.77866 - 314.577)}{\ln\left[\frac{(331.95418 - 301.09)}{(333.77866 - 314.577)}\right]} = 24.573K$$

The temperature correction factor as given by equation (3.4.10) is

$$F_T = 1 - \sum_{i=1}^{4}\sum_{k=1}^{4} a_{i,k}(1-\phi_3)^k \sin\left[2i \arctan(\phi_1 / \phi_2)\right] \qquad \text{(a)}$$

where $\phi_1 = \frac{T_{wi} - T_{wo}}{T_{wi} - T_{ai}} = \frac{60.62866 - 58.80418}{60.62866 - 27.94} = 0.05581$

$$\phi_2 = \frac{T_{ao} - T_{ai}}{T_{wi} - T_{ai}} = \frac{41.427 - 27.94}{60.62866 - 27.94} = 0.4126$$

$$\phi_3 = \frac{\phi_1 - \phi_2}{\ln\left[\frac{(1-\phi_2)}{(1-\phi_1)}\right]} = \frac{0.05581 - 0.4126}{\ln\left[\frac{(1-0.4126)}{(1-0.05581)}\right]} = 0.75176$$

Substituting into equation (a) above, one obtains,

$$F_T = 1 - \sum_{i=1}^{4}\sum_{k=1}^{4} a_{i,k}(0.2482)^k \sin(2i(7.7033))$$

For crossflow with one tube row [98KR1] (using Table B.2)

Table B.2: Sixteen values of the empirical constant $a_{i,k}$.

| $F_T = 1 - \Sigma$ | i = 1 | i = 2 | i = 3 | i = 4 | |
|---|---|---|---|---|---|
| | -0.03046 | -0.00398 | -0.03118 | -0.00917 | k = 1 |
| | 0.08314 | 0.01669 | 0.05871 | 0.01881 | k = 2 |
| | -0.06377 | -0.01856 | -0.03235 | -0.01147 | k = 3 |
| | 0.01734 | 0.006182 | 0.00453 | 0.002167 | k = 4 |
| | 0.00625 | 0.006182 | 0.000633 | 0.000337 | **0.007552** |

therefore $F_T = 1 - 0.007552 = 0.992448$

Upon substitution of equation (3.4.7) into equation (3.4.6) find

$$N_y = \left\{ \left[ \frac{F_T \Delta T_{lm}}{Q_a} - \frac{1}{h_w A_w} \right] A_{fr} k_{av} \, \mathrm{Pr}_{av}^{0.333} \right\}^{-1}$$

$$= \left\{ \left[ \frac{0.992448x24.573}{15997.404} - (10404.68x0.96464)^{-1} \right] x2.6599x10^{-2} \, x0.71761^{0.333} \, x0.235 \right\}^{-1}$$

$$= 125398.862 \ \mathrm{m}^{-1}$$

The corresponding characteristic flow parameter is according to equation (3.4.5)

$$Ry = \frac{m_a}{\mu_{av} A_{fr}} = \frac{1.155326}{1.859x10^{-5} \, x0.235} = 2.6446x10^5 \, m^{-1}$$

# APPENDIX C

## Thermal-flow performance of C-2.5 and C-4.3 elliptical finned tubes

## Table C.1.1: Tube specifications.

**Name: C-2.5** (2.5 mm fin pitch, 0.6676 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 4.63 | 188 | 468 | 2.48936 |
| 2 | 4.7 | 191 | 470 | 2.46073 |
| 3 | 4.69 | 188 | 468 | 2.48936 |
| 4 | 4.72 | 188 | 468 | 2.48936 |
| 5 | 4.67 | 191 | 467 | 2.44503 |
| 6 | 4.79 | 189 | 468 | 2.47619 |
| 7 | 4.835 | 193 | 480 | 2.48705 |
| 8 | 4.69 | 189 | 469 | 2.48148 |
| 9 | 4.76 | 189 | 468 | 2.47619 |
| 10 | 4.735 | 189 | 470 | 2.48677 |
| Average | 4.722 | 189.5 | 469.6 | 2.4781526 |

## Table C.1.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 22.63694 | 35.46069 | 60.88855 | 58.01786 | 100450.27 | 2.4114289 | 2.5673188 |
| 2 | 22.74278 | 37.00472 | 60.97183 | 58.49038 | 100448.55 | 1.9983866 | 2.7362994 |
| 3 | 22.70082 | 39.07977 | 60.9907 | 58.78043 | 100449.23 | 1.5429923 | 2.7319314 |
| 4 | 22.34937 | 41.40487 | 61.09417 | 59.00447 | 100454.97 | 1.1567169 | 2.5795728 |
| 5 | 22.0392 | 44.14231 | 61.17526 | 59.39847 | 100460.04 | 0.854929 | 2.5767677 |
| 6 | 22.15795 | 46.48541 | 61.24638 | 59.69843 | 100458.1 | 0.6689474 | 2.5791408 |

## Table C.1.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 547 | 404 | 1354 |
| 2 | 401 | 750 | 958 |
| 3 | 260 | 447 | 592 |
| 4 | 162 | 252 | 350 |
| 5 | 100 | 139 | 202 |
| 6 | 68 | 86 | 132 |

## Table C.1.3: Pressure drops for isothermal tests.

($p_{atm}$ = 101430 N/m$^2$, $T_{aiwb}$ = 15°C, $T_{aidb}$ = 19°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 564 | 433 | 1425 |
| 2 | 392 | 755 | 954 |
| 3 | 250 | 443 | 580 |
| 4 | 156 | 245 | 347 |
| 5 | 101 | 150 | 213 |
| 6 | 61 | 79 | 119 |

**Table C.1.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 30.13090638 | 0.9564751 | 31442.12 | 30834.09 | 0.9806619 | 1.9338069 |
| 2 | 29.46590932 | 0.9608632 | 28978.869 | 28408.779 | 0.9803274 | 1.9672594 |
| 3 | 28.4088203 | 0.9630261 | 25698.099 | 25264.392 | 0.983123 | 1.6877041 |
| 4 | 27.29914972 | 0.9634797 | 22416.793 | 22554.668 | 1.0061505 | -0.615053 |
| 5 | 25.87918025 | 0.9679135 | 19221.341 | 19157.203 | 0.9966632 | 0.3336818 |
| 6 | 24.40400448 | 0.9708886 | 16553.842 | 16705.725 | 1.0091751 | -0.917508 |

**Table C.1.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 107119.3103 | 11072.746 | 556126.26 | 220160.37 | 11.949575 |
| 2 | 114658.2384 | 11702.131 | 459919.53 | 203468.03 | 12.720941 |
| 3 | 114746.4968 | 11697.439 | 354227.47 | 184690.82 | 13.788712 |
| 4 | 108618.9239 | 11157.846 | 264928.35 | 166386.9 | 15.237687 |
| 5 | 108895.0306 | 11163.784 | 195241.93 | 148034.52 | 17.14994 |
| 6 | 109303.8378 | 11185.002 | 152307.21 | 133392.27 | 18.971854 |

**Correlating equations:** $Ny = 1413.3189Ry^{0.381476}$

$K_{he} = 1295.972Ry^{-0.354874}$

**Table C.1.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 608310.4794 | 11.305539 |
| 2 | 490022.2449 | 12.10925 |
| 3 | 376688.2366 | 13.068888 |
| 4 | 280759.9983 | 14.67969 |
| 5 | 219739.4802 | 15.515577 |
| 6 | 159188.9651 | 17.85528 |

**Correlating equation:** $K_{heiso} = 979.4976Ry^{-0.335451}$

## Table C.2.1: Tube specifications.

**Name: C-4.3** (4.3 mm fin pitch, 0.62383 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 3.24 | 109 | 463 | 4.24771 |
| 2 | 3.235 | 110 | 466 | 4.23636 |
| 3 | 3.26 | 110 | 466 | 4.23636 |
| 4 | 3.29 | 111 | 470 | 4.23423 |
| 5 | 3.23 | 110 | 470 | 4.27273 |
| 6 | 3.25 | 111 | 470 | 4.23423 |
| 7 | 3.225 | 109 | 470 | 4.31193 |
| 8 | 3.23 | 109 | 471 | 4.32110 |
| 9 | 3.285 | 111 | 471 | 4.24324 |
| 10 | 3.245 | 109 | 470 | 4.31193 |
| Average | 3.249 | 109.9 | 468.7 | 4.2649827 |

## Table C.2.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 23.90791 | 32.15374 | 62.732685 | 60.740225 | 100608.82 | 2.6455679 | 2.6408696 |
| 2 | 24.43415 | 33.67472 | 62.557835 | 60.789945 | 100600.21 | 2.0239515 | 2.5962942 |
| 3 | 23.41977 | 34.48564 | 62.702545 | 61.098195 | 100616.81 | 1.5587699 | 2.5659763 |
| 4 | 23.54864 | 36.54361 | 62.888465 | 61.579035 | 100614.7 | 1.153519 | 2.7679944 |
| 5 | 23.29122 | 38.31861 | 63.186735 | 62.030035 | 100618.91 | 0.8917813 | 2.7248441 |
| 6 | 23.63806 | 41.08453 | 63.224555 | 62.278015 | 100613.23 | 0.6263262 | 2.7080537 |

## Table C.2.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 288 | 480 | 1250 |
| 2 | 179 | 758 | 746 |
| 3 | 114 | 448 | 450 |
| 4 | 69 | 246 | 253 |
| 5 | 45 | 148 | 156 |
| 6 | 25 | 74 | 81 |

## Table C.2.3: Pressure drops for isothermal tests.

($p_{atm}$ = 100750 N/m$^2$, $T_{aiwb}$ = 21°C, $T_{aidb}$ = 26°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 278 | 485 | 1278 |
| 2 | 170 | 743 | 740 |
| 3 | 109 | 456 | 462 |
| 4 | 68 | 252 | 262 |
| 5 | 44 | 152 | 160 |
| 6 | 26 | 79 | 86 |

**Table C.2.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 33.60872537 | 0.9823922 | 22169.363 | 22022.052 | 0.9933552 | 0.6644799 |
| 2 | 32.47629485 | 0.9825484 | 19003.755 | 19209.937 | 1.0108495 | -1.084951 |
| 3 | 32.71998598 | 0.9824204 | 17533.16 | 17229.974 | 0.9827078 | 1.729215 |
| 4 | 31.83093291 | 0.9835946 | 15236.864 | 15170.651 | 0.9956544 | 0.4345555 |
| 5 | 31.29278531 | 0.9840678 | 13623.623 | 13193.08 | 0.9683973 | 3.1602694 |
| 6 | 29.62818508 | 0.9854135 | 11108.093 | 10729.771 | 0.9659417 | 3.4058294 |

**Table C.2.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 114079.6738 | 11496.069 | 611476.7 | 129856.32 | 5.254843 |
| 2 | 112048.7172 | 11328.741 | 466557.47 | 114108.19 | 5.5616406 |
| 3 | 111117.5092 | 11232.655 | 359500.24 | 104086.49 | 5.9731223 |
| 4 | 120465.5305 | 11993.992 | 265310.77 | 91786.089 | 6.5780278 |
| 5 | 119252.206 | 11862.733 | 204739.91 | 82969.924 | 7.1597403 |
| 6 | 118769.6297 | 11811.16 | 143235.51 | 70657.235 | 8.022957 |

**Correlating equations:** $Ny = 536.9496Ry^{0.411508}$

$K_{he} = 269.315Ry^{-0.296708}$

**Table C.2.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 623628.3068 | 4.9708387 |
| 2 | 471204.8223 | 5.3243425 |
| 3 | 370234.2651 | 5.5298047 |
| 4 | 275810.3367 | 6.216191 |
| 5 | 214200.2188 | 6.6688257 |
| 6 | 154119.4907 | 7.611929 |

**Correlating equation:** $K_{heiso} = 282.3795Ry^{-0.304296}$

**Table C.3.1a: Test data.**

**Name: B-4.3 & B-2.5** (4.3 mm and 2.5 mm fin pitch)

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 24.01777 | 42.8982 | 60.79502 | 57.95466 | 100198.62 | 2.2342124 | 3.6239146 |
| 2 | 24.5186 | 44.52581 | 60.79723 | 58.2151 | 100190.46 | 1.9290934 | 3.603546 |
| 3 | 25.01257 | 46.95918 | 61.33446 | 59.05827 | 100182.41 | 1.5348638 | 3.5825708 |
| 4 | 25.08054 | 49.90852 | 61.77853 | 59.86683 | 100181.3 | 1.1351751 | 3.5750793 |
| 5 | 24.94244 | 52.51865 | 62.50963 | 60.92291 | 100183.55 | 0.8812734 | 3.5495136 |
| 6 | 25.20503 | 55.31199 | 62.89511 | 61.64023 | 100179.27 | 0.6432988 | 3.5717599 |

**Table C.3.1b: Test data.**

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 688 | 356 | 1394 |
| 2 | 540 | 718 | 1072 |
| 3 | 366 | 455 | 706 |
| 4 | 226 | 250 | 411 |
| 5 | 150 | 152 | 262 |
| 6 | 91.5 | 82 | 152 |

**Table C.3.2: Pressure drops for isothermal tests.**

($p_{atm}$ = 101070 N/m$^2$, $T_{aiwb}$ = 17°C, $T_{aidb}$ = 21°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 685 | 397 | 1472 |
| 2 | 503 | 742 | 1050 |
| 3 | 332 | 456 | 670 |
| 4 | 200 | 250 | 385 |
| 5 | 129 | 147 | 238 |
| 6 | 77 | 78 | 136 |

**Table C.3.3a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 25.067326 | 0.9949094 | 42879.931 | 43063.689 | 1.0042854 | -0.428541 |
| 2 | 23.936083 | 0.9949537 | 39228.798 | 38929.31 | 0.9923656 | 0.7634386 |
| 3 | 22.814387 | 0.995272 | 34234.239 | 34120.698 | 0.9966834 | 0.3316595 |
| 4 | 21.31334 | 0.9957998 | 28645.268 | 28599.785 | 0.9984122 | 0.1587783 |
| 5 | 20.28379 | 0.9962498 | 24702.737 | 23571.593 | 0.9542098 | 4.5790245 |
| 6 | 18.38168 | 0.9962257 | 19686.894 | 18760.465 | 0.9529418 | 4.7058161 |

**Table C.3.3b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 151022.7 | 14628.276 | 509549 | 327665.32 | 17.219154 |
| 2 | 150477.55 | 14570.47 | 438772.45 | 312218.9 | 18.066432 |
| 3 | 151191.7 | 14558.944 | 347829.07 | 283141.22 | 19.252307 |
| 4 | 152320.06 | 14587.24 | 256314.13 | 250911.54 | 21.627792 |
| 5 | 153284.57 | 14575.26 | 198402.1 | 225438.13 | 23.723018 |
| 6 | 155524.92 | 14698.801 | 144290.34 | 196299.7 | 27.026035 |

**Correlating equations:**

$$Ny = 1552.2323Ry^{0.407909}$$

$$K_{he} = 1838.65Ry^{-0.356195}$$

**Table C.3.4: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 576611.2 | 14.977924 |
| 2 | 480713.44 | 15.824232 |
| 3 | 378150.49 | 16.878582 |
| 4 | 280711.59 | 18.451705 |
| 5 | 215303.11 | 20.230951 |
| 6 | 156560.52 | 22.837755 |

**Correlating equation:**

$$K_{heiso} = 1027.948Ry^{-0.319399}$$

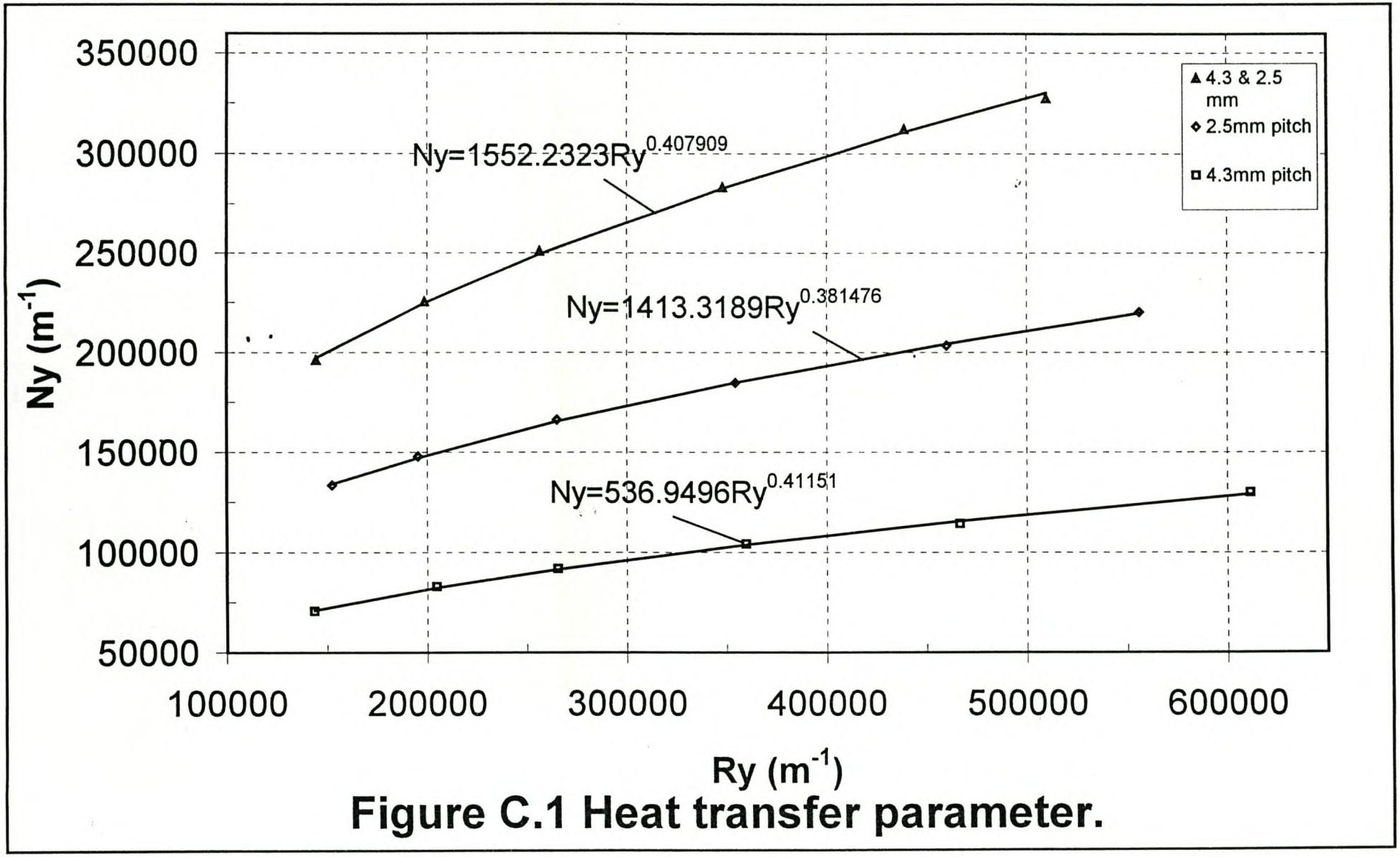



**Figure C.1 Heat transfer parameter.**

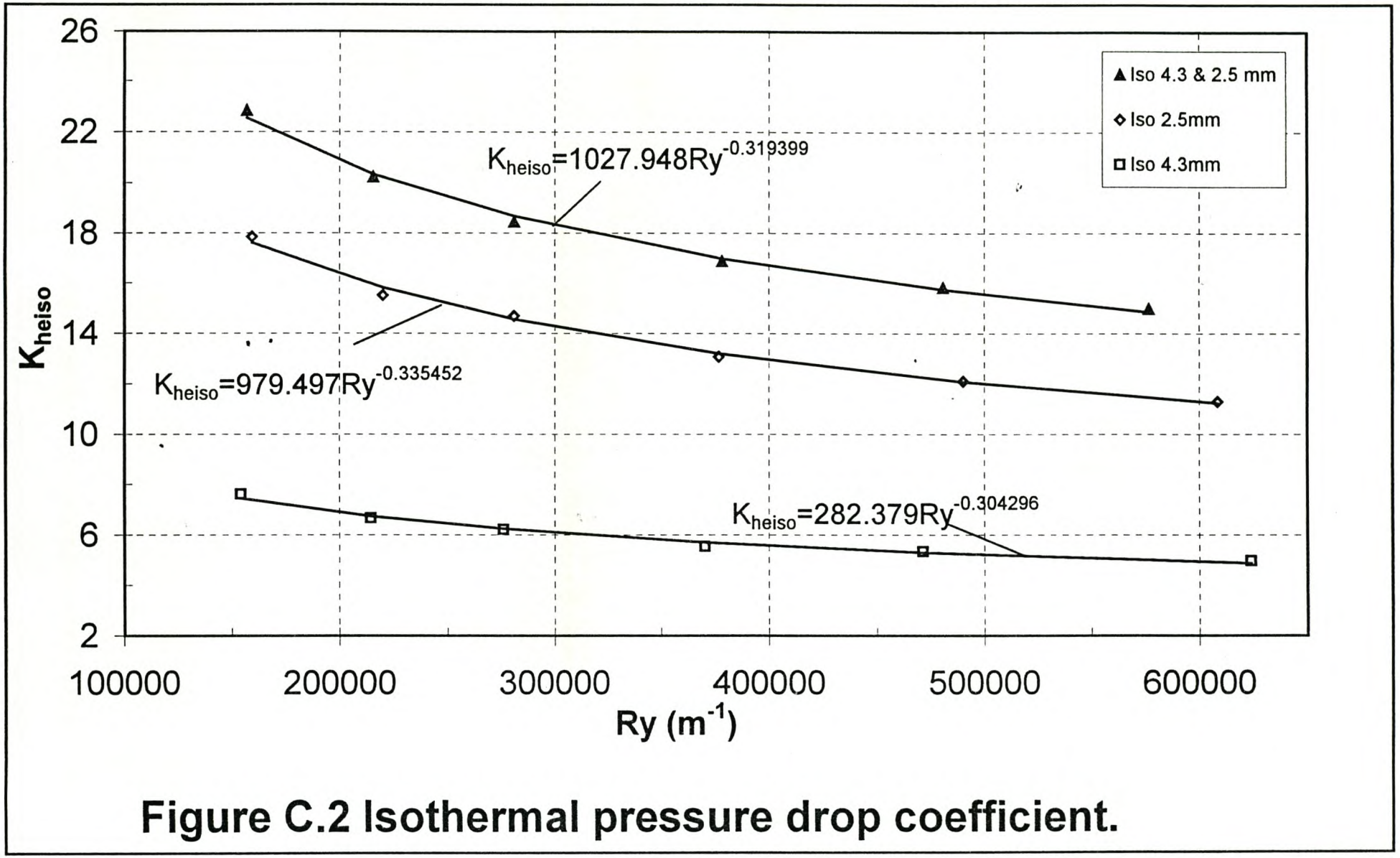



**Figure C.2 Isothermal pressure drop coefficient.**

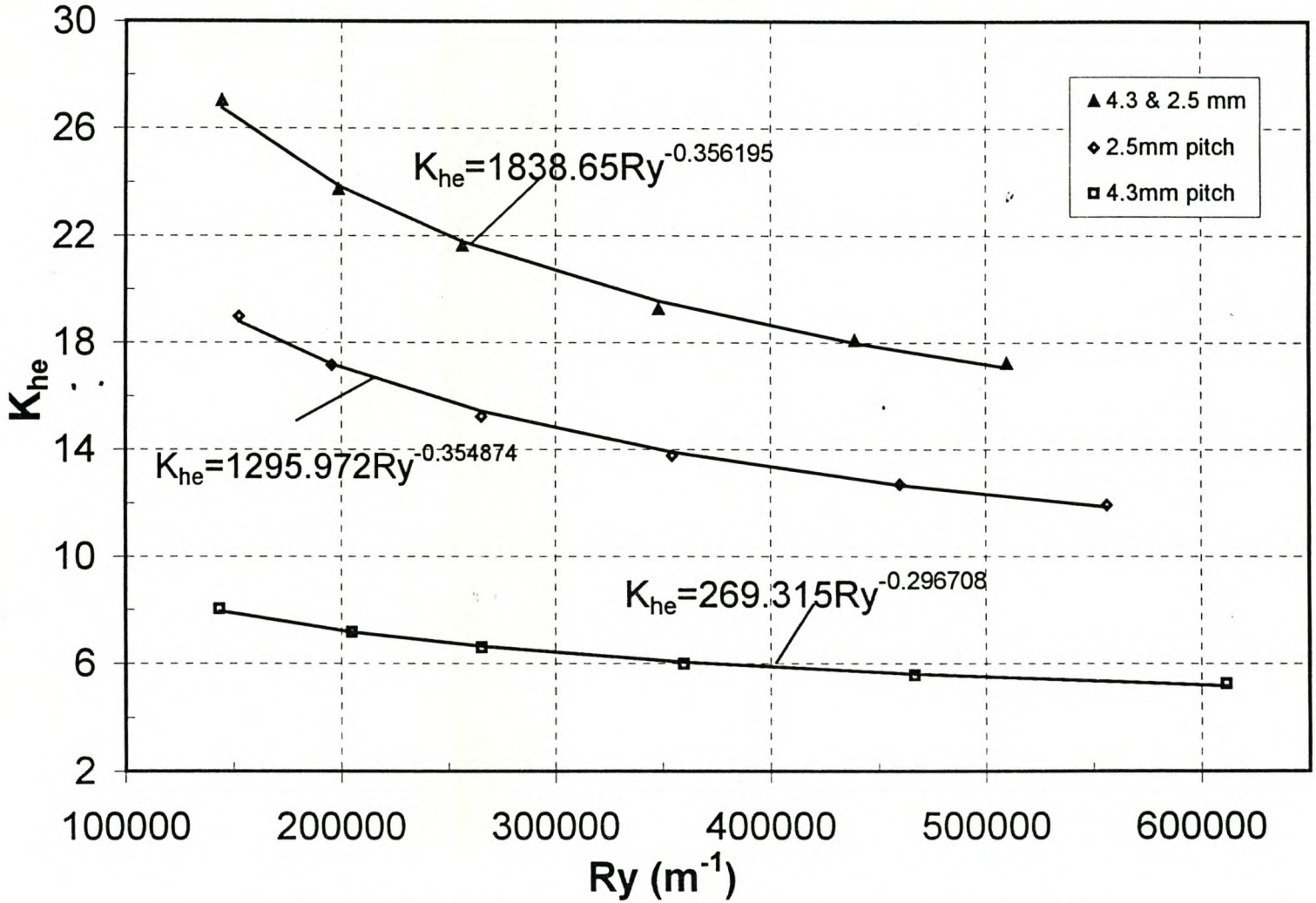



**Figure C.3 Non-isothermal pressure drop coefficient.**

# APPENDIX D

## Thermal-flow performance of AE elliptical finned tubes

## Table D.1.1: Tube specifications.

**Name: AE-2.5 DE** (2.5 mm fin pitch, 0.54373 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 4.515 | 188 | 472 | 2.5106 |
| 2 | 4.51 | 194 | 470 | 2.4227 |
| 3 | 4.515 | 193 | 470 | 2.4352 |
| 4 | 4.525 | 195 | 470 | 2.4103 |
| 5 | 4.515 | 193 | 467 | 2.4197 |
| 6 | 4.585 | 195 | 468 | 2.4000 |
| 7 | 4.645 | 195 | 470 | 2.4103 |
| 8 | 4.58 | 194 | 470 | 2.4227 |
| 9 | 4.665 | 196 | 470 | 2.3980 |
| 10 | 4.47 | 192 | 469 | 2.4427 |
| 11 | 4.56 | 191 | 470 | 2.4607 |
| 12 | 4.585 | 196 | 470 | 2.3980 |
| 13 | 4.53 | 193 | 469 | 2.4301 |
| 14 | 4.585 | 195 | 467 | 2.3949 |
| 15 | 4.58 | 193 | 470 | 2.4352 |
| 16 | 4.545 | 194 | 470 | 2.4227 |
| 17 | 4.515 | 192 | 469 | 2.4427 |
| 18 | 4.52 | 192 | 470 | 2.4479 |
| 19 | 4.535 | 194 | 469 | 2.4175 |
| 20 | 4.635 | 196 | 469 | 2.3929 |
| 21 | 4.445 | 192 | 466 | 2.4271 |
| 22 | 4.51 | 193 | 471 | 2.4404 |
| Average | 4.548636364 | 193.45455 | 469.36364 | 2.42646077 |

## Table D.1.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 18.000756 | 41.94349 | 60.049115 | 55.542005 | 101084.36 | 2.1918986 | 2.771560778 |
| 2 | 17.78178 | 43.43756 | 60.222875 | 56.009055 | 101087.96 | 1.8905878 | 2.728565636 |
| 3 | 18.253286 | 46.04914 | 60.281075 | 56.583625 | 101080.22 | 1.5120793 | 2.676699839 |
| 4 | 18.057246 | 49.05049 | 60.586235 | 57.399735 | 101083.44 | 1.145452 | 2.670944692 |
| 5 | 18.014246 | 51.4846 | 60.657525 | 58.011875 | 101084.14 | 0.8889927 | 2.755934015 |
| 6 | 17.76322 | 53.97822 | 60.694525 | 58.554575 | 101088.26 | 0.6589948 | 2.724975279 |

## Table D.1.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 834 | 339 | 1467 |
| 2 | 675.5 | 682 | 1164 |
| 3 | 468 | 437 | 781 |
| 4 | 303.5 | 252 | 488 |
| 5 | 206 | 153 | 318 |
| 6 | 130 | 85 | 182 |

**Table D.1.3: Pressure drops for isothermal tests.**

($p_{atm}$ = 101030 N/m$^2$, $T_{aiwb}$ = 14°C, $T_{aidb}$ = 14.8°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 895 | 369 | 1598 |
| 2 | 721 | 746 | 1257 |
| 3 | 466 | 443 | 788 |
| 4 | 288 | 244 | 467 |
| 5 | 184 | 141 | 287 |
| 6 | 126 | 87 | 190 |

**Table D.1.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 26.65270693 | 0.9921082 | 53390.416 | 52249.4479 | 0.9786297 | 2.1370271 |
| 2 | 26.05199401 | 0.9924196 | 49351.249 | 48093.8428 | 0.9745213 | 2.5478716 |
| 3 | 24.3233134 | 0.9928793 | 42759.475 | 41400.0653 | 0.968208 | 3.1792012 |
| 4 | 22.66506879 | 0.9931419 | 36122.946 | 35605.2306 | 0.985668 | 1.4332032 |
| 5 | 20.93267572 | 0.9930501 | 30278.329 | 30504.1172 | 1.0074571 | -0.745709 |
| 6 | 18.88932046 | 0.9919963 | 24288.959 | 24397.3442 | 1.0044623 | -0.446234 |

**Table D.1.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ W/m$^2$ K | Ry m$^{-1}$ | Ny m$^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 112705.0131 | 11558.43 | 504521.43 | 401259.989 | 22.115088 |
| 2 | 111513.4369 | 11430.366 | 434505.7 | 377015.779 | 24.025639 |
| 3 | 109933.999 | 11269.615 | 346163.17 | 346266.837 | 25.891566 |
| 4 | 110655.0516 | 11288.538 | 261345.05 | 310018.961 | 29.125196 |
| 5 | 114779.5945 | 11614.276 | 202244.48 | 277864.616 | 32.692161 |
| 6 | 113997.1768 | 11524.943 | 149522.25 | 244765.023 | 37.407998 |

**Correlating equations:** $Ny = 1983.295Ry^{0.404505}$

$K_{he} = 5811.972Ry^{-0.42394}$

**Table D.1.5: Isothermal test results.**

| Run no. | Ry m$^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 570669.2945 | 21.076518 |
| 2 | 494521.257 | 22.610501 |
| 3 | 382605.6777 | 24.413381 |
| 4 | 284718.1952 | 27.246259 |
| 5 | 216530.3702 | 30.097126 |
| 6 | 169890.4147 | 33.479398 |

**Correlating equation:** $K_{heiso} = 2891.203Ry^{-0.37098}$

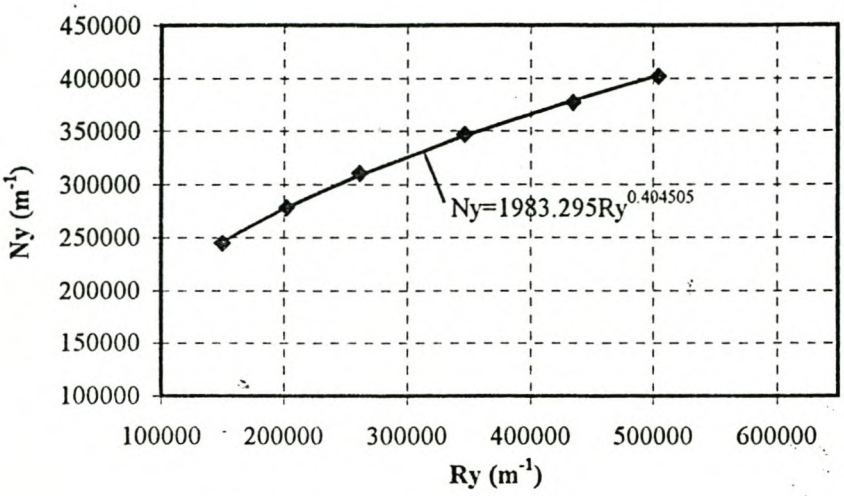
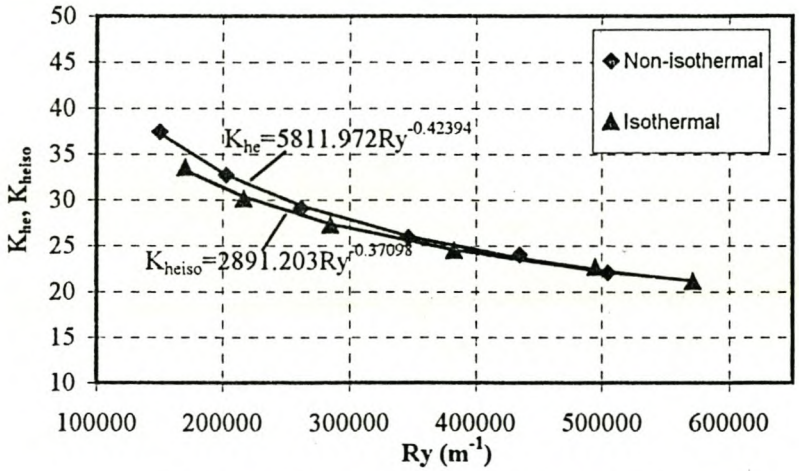



**Figure D.1.1: Heat transfer parameter ( 2.5mm pitch Double-Row).**




**D.1.2: Pressure drop coefficient (2.5mm pitch Double row).**

## Table D.2.1: Tube specifications.

**Name: AE-4.0 DE** (4.0 mm fin pitch, 0.60250 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 3.36 | 116 | 469 | 4.04310 |
| 2 | 3.355 | 116 | 468 | 4.03448 |
| 3 | 3.335 | 115 | 465 | 4.04348 |
| 4 | 3.365 | 115 | 469 | 4.07826 |
| 5 | 3.39 | 116 | 469 | 4.04310 |
| 6 | 3.375 | 116 | 468 | 4.03448 |
| 7 | 3.36 | 116 | 469 | 4.04310 |
| 8 | 3.385 | 116 | 468 | 4.03448 |
| 9 | 3.34 | 116 | 469 | 4.04310 |
| 10 | 3.39 | 116 | 469 | 4.04310 |
| 11 | 3.355 | 116 | 470 | 4.05172 |
| 12 | 3.37 | 114 | 469 | 4.11404 |
| 13 | 3.37 | 116 | 470 | 4.05172 |
| 14 | 3.375 | 116 | 470 | 4.05172 |
| 15 | 3.35 | 115 | 466 | 4.05217 |
| 16 | 3.375 | 116 | 466 | 4.01724 |
| 17 | 3.36 | 116 | 468 | 4.03448 |
| 18 | 3.415 | 118 | 471 | 3.99153 |
| 19 | 3.37 | 116 | 467 | 4.02586 |
| 20 | 3.345 | 117 | 469 | 4.00855 |
| 21 | 3.355 | 116 | 468 | 4.03448 |
| 22 | 3.36 | 116 | 468 | 4.03448 |
| Average | 3.366136364 | 115.90909 | 468.40909 | 4.04130501 |

## Table D.2.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 22.19607 | 37.61024 | 60.70259 | 57.4983 | 100447.51 | 2.5153157 | 2.757777389 |
| 2 | 22.10328 | 39.47038 | 60.82777 | 57.97027 | 100449.03 | 1.9813804 | 2.724005892 |
| 3 | 22.34393 | 41.56862 | 60.88971 | 58.35859 | 100445.1 | 1.5585997 | 2.725377256 |
| 4 | 22.59647 | 44.07004 | 60.90774 | 58.7297 | 100440.97 | 1.1469529 | 2.71334254 |
| 5 | 22.35981 | 46.19174 | 60.96447 | 59.03818 | 100444.84 | 0.8968511 | 2.710278721 |
| 6 | 22.31902 | 49.02559 | 61.06252 | 59.4862 | 100445.5 | 0.6502227 | 2.70559338 |

## Table D.2.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 476.5 | 443 | 1313 |
| 2 | 324 | 743 | 856 |
| 3 | 217 | 460 | 550 |
| 4 | 132 | 250 | 315 |
| 5 | 88.5 | 154 | 202 |
| 6 | 53 | 82 | 114 |

**Table D.2.3: Pressure drops for isothermal tests.**

($p_{atm}$ = 100840 N/m$^2$, $T_{aiwb}$ = 18 °C, $T_{aidb}$ = 20 °C)

| Run no. | $Dp_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 493 | 472 | 1384 |
| 2 | 324 | 760 | 871 |
| 3 | 206 | 442 | 527 |
| 4 | 127 | 246 | 307 |
| 5 | 84 | 149 | 194 |
| 6 | 51.5 | 81 | 109 |

**Table D.2.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 28.76671289 | 0.9970376 | 39472.107 | 36968.5979 | 0.9365752 | 6.3424764 |
| 2 | 27.98814159 | 0.9962921 | 35035.322 | 32565.3508 | 0.9295005 | 7.0499451 |
| 3 | 26.80712556 | 0.9959376 | 30506.62 | 28861.2872 | 0.9460664 | 5.3933629 |
| 4 | 25.26940676 | 0.9959642 | 25075.1 | 24726.3346 | 0.9860912 | 1.3908843 |
| 5 | 24.08790033 | 0.9962174 | 21763.585 | 21844.2484 | 1.0037064 | -0.370636 |
| 6 | 22.28952395 | 0.9966965 | 17683.596 | 17845.4211 | 1.0091512 | -0.915116 |

**Table D.2.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ W/m$^2$ K | Ry m$^{-1}$ | Ny m$^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 114442.2599 | 11604.064 | 579338.22 | 265110.644 | 9.5319021 |
| 2 | 113562.2411 | 11505.578 | 455376.58 | 240198.43 | 10.414614 |
| 3 | 114013.293 | 11526.344 | 357162 | 216532.894 | 11.229621 |
| 4 | 113849.0981 | 11497.202 | 261931.63 | 186812.723 | 12.557716 |
| 5 | 114038.8833 | 11499.073 | 204356.96 | 168899.551 | 13.727409 |
| 6 | 114317.4415 | 11501.406 | 147660.3 | 146904.476 | 15.569253 |

**Correlating equations:** $Ny = 823.0402Ry^{0.435418}$

$K_{he} = 1068.022Ry^{-0.35579}$

**Table D.2.5: Isothermal test results.**

| Run no. | Ry m$^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 631030.3873 | 9.0667601 |
| 2 | 488675.764 | 9.9359446 |
| 3 | 373955.2601 | 10.78782 |
| 4 | 279588.8366 | 11.897886 |
| 5 | 217616.1529 | 12.989809 |
| 6 | 160178.6806 | 14.699534 |

**Correlating equation:** $K_{heiso} = 928.4973Ry^{-0.34682}$

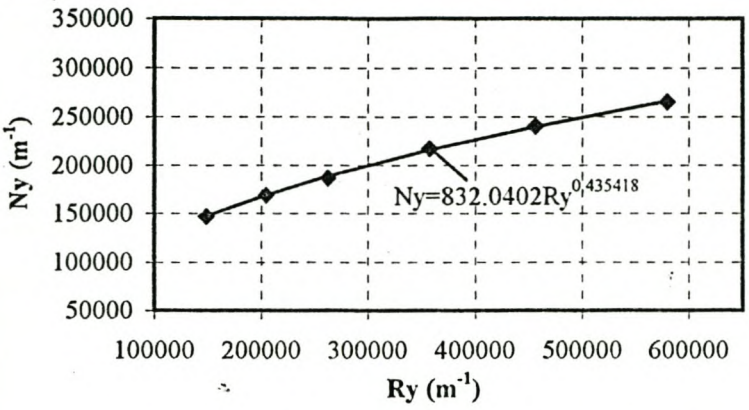



**Figure D.2.1: Heat transfer parameter (4.0mm pitch Double-Row).**

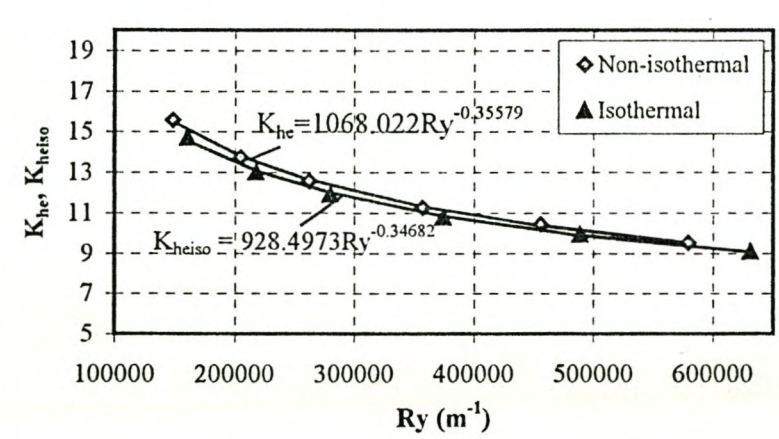



**Figure D.2.2: Pressure drop coefficient (4.0mm pitch Double row).**

## Table D.3.1: Tube specifications.

**Name: AE-4.3 DA** (4.3 mm fin pitch, 0.59120 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 3.155 | 110 | 468 | 4.25455 |
| 2 | 3.13 | 109 | 468 | 4.29358 |
| 3 | 3.125 | 109 | 469 | 4.30275 |
| 4 | 3.15 | 109 | 469 | 4.30275 |
| 5 | 3.13 | 109 | 469 | 4.30275 |
| 6 | 3.18 | 111 | 469 | 4.22523 |
| 7 | 3.155 | 109 | 470 | 4.31193 |
| 8 | 3.14 | 110 | 465 | 4.22727 |
| 9 | 3.135 | 108 | 467 | 4.32407 |
| 10 | 3.15 | 109 | 469 | 4.30275 |
| 11 | 3.115 | 108 | 469 | 4.34259 |
| 12 | 3.17 | 110 | 468 | 4.25455 |
| 13 | 3.185 | 111 | 469 | 4.22523 |
| 14 | 3.15 | 109 | 468 | 4.29358 |
| 15 | 3.115 | 112 | 470 | 4.19643 |
| 16 | 3.16 | 111 | 467 | 4.20721 |
| 17 | 3.15 | 108 | 469 | 4.34259 |
| 18 | 3.16 | 110 | 468 | 4.25455 |
| 19 | 3.17 | 110 | 469 | 4.26364 |
| 20 | 3.18 | 110 | 468 | 4.25455 |
| 21 | 3.17 | 111 | 467 | 4.20721 |
| 22 | 3.135 | 108 | 470 | 4.35185 |
| Average | 3.150454545 | 109.59091 | 468.40909 | 4.2746176 |

## Table D.3.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 22.15542 | 37.99629 | 60.8552 | 57.58429 | 100428.25 | 2.4815448 | 2.77640066 |
| 2 | 22.47405 | 39.77207 | 60.94714 | 57.99116 | 100423.04 | 1.9887458 | 2.768342594 |
| 3 | 22.33467 | 41.95096 | 61.01423 | 58.38749 | 100425.32 | 1.5402497 | 2.745846812 |
| 4 | 22.60989 | 44.40295 | 61.04463 | 58.7831 | 100420.83 | 1.1641088 | 2.711969902 |
| 5 | 22.42287 | 46.795 | 61.12537 | 59.20583 | 100423.88 | 0.8660041 | 2.686798435 |
| 6 | 22.42957 | 49.06822 | 61.1892 | 59.5537 | 100423.77 | 0.6618538 | 2.674419377 |

## Table D.3.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 509 | 432 | 1329 |
| 2 | 351 | 750 | 891 |
| 3 | 225 | 450 | 551 |
| 4 | 142 | 258 | 332 |
| 5 | 88 | 144 | 195 |
| 6 | 57 | 85 | 120 |

**Table D.3.3: Pressure drops for isothermal tests.**

($p_{atm}$ = 100810 N/m$^2$, $T_{aiwb}$ = 17.5 °C, $T_{aidb}$ = 21°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 528 | 462 | 1393 |
| 2 | 343 | 740 | 875 |
| 3 | 222 | 450 | 546 |
| 4 | 138.5 | 258 | 328 |
| 5 | 90 | 154 | 203 |
| 6 | 49 | 74 | 103 |

**Table D.3.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 28.68635374 | 0.9966653 | 40043.456 | 37992.727 | 0.9487874 | 5.1212582 |
| 2 | 27.73069827 | 0.9960661 | 35041.333 | 34236.347 | 0.9770275 | 2.2972461 |
| 3 | 26.66191632 | 0.9956645 | 30779.185 | 30176.9133 | 0.9804325 | 1.9567494 |
| 4 | 25.15625804 | 0.9957708 | 25843.436 | 25661.5474 | 0.9929619 | 0.7038104 |
| 5 | 23.8184941 | 0.9961961 | 21503.367 | 21579.6695 | 1.0035484 | -0.354838 |
| 6 | 22.33761717 | 0.9965787 | 17963.618 | 18302.3363 | 1.0188558 | -1.885578 |

**Table D.3.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ W/m$^2$ K | Ry m$^{-1}$ | Ny m$^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 115427.3457 | 11678.509 | 571534.74 | 269899.528 | 10.448912 |
| 2 | 115535.1961 | 11667.853 | 456821.99 | 242254.559 | 11.180471 |
| 3 | 115004.9345 | 11604.521 | 352927.74 | 219720.849 | 11.90841 |
| 4 | 113957.4399 | 11498.98 | 265835.25 | 193689.742 | 13.098725 |
| 5 | 113335.0635 | 11426.69 | 197238.76 | 168673.613 | 14.615221 |
| 6 | 113167.6295 | 11396.643 | 150325.69 | 149038.025 | 16.147594 |

**Correlating equations:** $Ny = 778.019Ry^{0.441216}$

$K_{he} = 770.0977Ry^{-0.32518}$

**Table D.3.5: Isothermal test results.**

| Run no. | Ry m$^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 621343.6689 | 9.9204324 |
| 2 | 479950.011 | 10.800965 |
| 3 | 375477.1724 | 11.42211 |
| 4 | 284917.2478 | 12.375784 |
| 5 | 220185.9277 | 13.465542 |
| 6 | 152313.3386 | 15.320772 |

**Correlating equation:** $K_{heiso} = 565.441Ry^{-0.30335}$

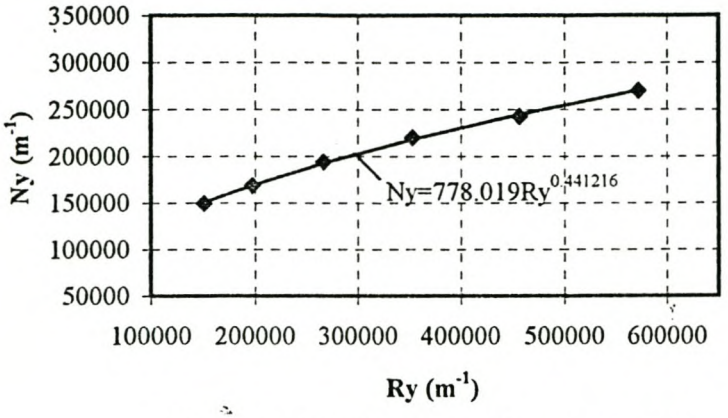



**Figure D.3.1: Heat transfer parameter (4.3mm pitch Double-Row).**

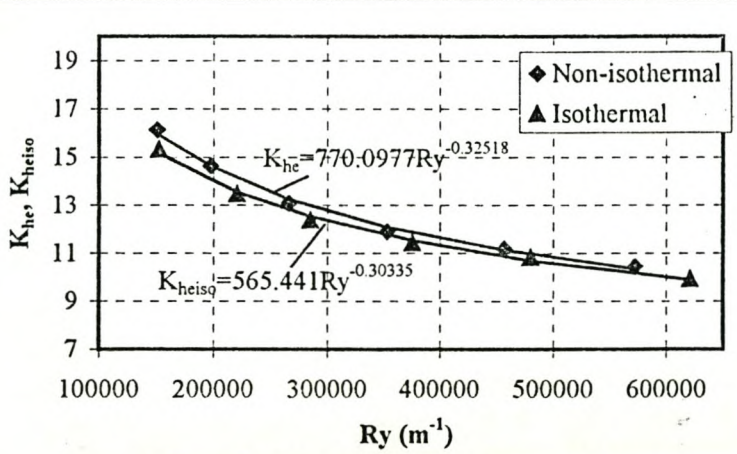



**Figure D.3.2 Pressure drop coefficient (4.3mm pitch Double row).**

## Table D.4.1: Tube specifications.

Name: **AE-4.5 DA** (4.5 mm fin pitch, 0.59020 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 3.155 | 107 | 471 | 4.40187 |
| 2 | 3.15 | 108 | 470 | 4.35185 |
| 3 | 3.145 | 107 | 469 | 4.38318 |
| 4 | 3.135 | 106 | 467 | 4.40566 |
| 5 | 3.14 | 106 | 470 | 4.43396 |
| 6 | 3.155 | 106 | 466 | 4.39623 |
| 7 | 3.135 | 106 | 465 | 4.38679 |
| 8 | 3.125 | 106 | 465 | 4.38679 |
| 9 | 3.135 | 107 | 468 | 4.37383 |
| 10 | 3.145 | 106 | 467 | 4.40566 |
| 11 | 3.13 | 106 | 466 | 4.39623 |
| 12 | 3.15 | 106 | 470 | 4.43396 |
| 13 | 3.15 | 106 | 467 | 4.40566 |
| 14 | 3.115 | 105 | 468 | 4.45714 |
| 15 | 3.16 | 107 | 469 | 4.38318 |
| 16 | 3.18 | 108 | 473 | 4.37963 |
| 17 | 3.13 | 106 | 470 | 4.43396 |
| 18 | 3.14 | 109 | 466 | 4.27523 |
| 19 | 3.13 | 106 | 467 | 4.40566 |
| 20 | 3.14 | 108 | 471 | 4.36111 |
| 21 | 3.1 | 105 | 470 | 4.47619 |
| 22 | 3.145 | 106 | 469 | 4.42453 |
| Average | 3.140454545 | 106.5 | 468.36364 | 4.3981048 |

## Table D.4.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 24.55259 | 38.70659 | 60.890515 | 57.854315 | 100269.58 | 2.5187336 | 2.793056647 |
| 2 | 24.51718 | 40.46827 | 60.974365 | 58.250885 | 100270.16 | 1.9699321 | 2.770165948 |
| 3 | 24.10778 | 42.28182 | 60.992925 | 58.584095 | 100276.84 | 1.5328636 | 2.740891816 |
| 4 | 23.4151 | 44.42199 | 61.079555 | 58.968735 | 100288.13 | 1.1368095 | 2.738389246 |
| 5 | 23.11823 | 46.36214 | 61.093505 | 59.244675 | 100292.98 | 0.8904905 | 2.729749142 |
| 6 | 23.02898 | 49.17038 | 61.224655 | 59.742825 | 100294.43 | 0.6337169 | 2.717479639 |

## Table D.4.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 457 | 446 | 1312 |
| 2 | 308.5 | 737 | 844 |
| 3 | 201 | 446 | 526 |
| 4 | 124 | 246 | 306 |
| 5 | 83 | 152 | 195 |
| 6 | 49 | 78 | 106 |

**Table D.4.3: Pressure drops for isothermal tests.**

($p_{atm}$ = 100680 N/m$^2$, $T_{aiwb}$ = 17 °C, $T_{aidb}$ = 20.5 °C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 479 | 475 | 1374 |
| 2 | 308 | 752 | 857 |
| 3 | 196 | 446 | 522 |
| 4 | 123 | 255 | 312 |
| 5 | 81.5 | 155 | 197 |
| 6 | 44 | 73 | 98 |

**Table D.4.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 27.36748042 | 0.9973872 | 36228.781 | 35478.8759 | 0.9793008 | 2.069915 |
| 2 | 26.5734545 | 0.996492 | 31934.341 | 31564.9821 | 0.9884338 | 1.1566185 |
| 3 | 25.7957482 | 0.996049 | 28317.128 | 27623.9194 | 0.9755198 | 2.4480189 |
| 4 | 24.92300179 | 0.996044 | 24280.946 | 24185.1839 | 0.9960561 | 0.394393 |
| 5 | 23.85058738 | 0.9963319 | 21048.385 | 21117.0131 | 1.0032605 | -0.32605 |
| 6 | 22.14133494 | 0.9968762 | 16848.058 | 16849.9569 | 1.0001127 | -0.011269 |

**Table D.4.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ W/m$^2$ K | Ry m$^{-1}$ | Ny m$^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 116393.2542 | 11748.412 | 576938.37 | 253576.138 | 9.0612569 |
| 2 | 115866.4748 | 11684.569 | 450280.97 | 228625.732 | 9.9714874 |
| 3 | 114951.8051 | 11593.114 | 349809.94 | 207609.198 | 10.70497 |
| 4 | 115261.8899 | 11600.844 | 259012.11 | 182822.2 | 11.978254 |
| 5 | 115152.9544 | 11580.254 | 202499.59 | 164493.401 | 13.031539 |
| 6 | 115186.4964 | 11558.469 | 143637.93 | 140437.453 | 15.123945 |

**Correlating equations:** $Ny = 941.0388Ry^{0.422242}$

$K_{he} = 1087.139Ry^{-0.36117}$

**Table D.4.5: Isothermal test results.**

| Run no. | Ry m$^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 630926.5652 | 8.7532727 |
| 2 | 484519.1784 | 9.5438004 |
| 3 | 374385.1311 | 10.17212 |
| 4 | 283683.6542 | 11.118061 |
| 5 | 221224.1211 | 12.113935 |
| 6 | 151491.9293 | 13.946533 |

**Correlating equation:** $K_{heiso} = 641.4488Ry^{-0.32206}$

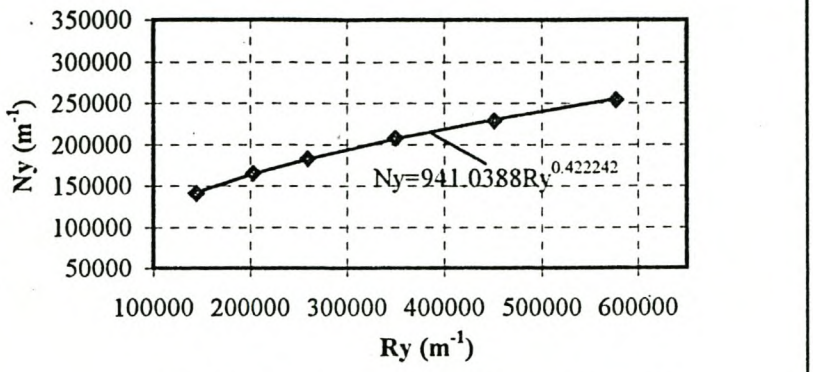



**Figure D.4.1: Heat transfer parameter (4.5mm pitch Double-Row).**

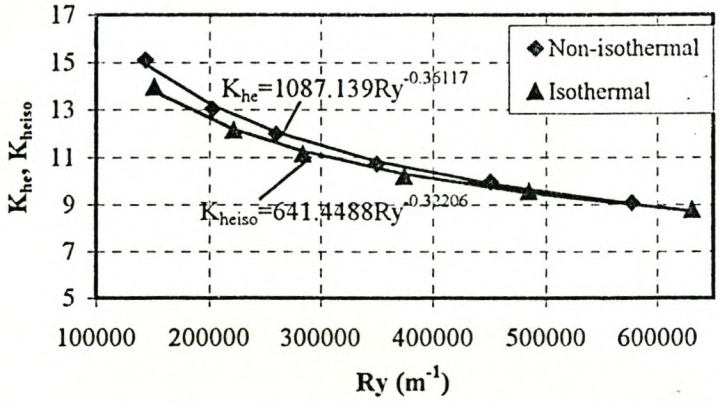



**Figure D.4.2: Pressure drop coefficient (4.5mm pitch Double row).**

**Table D.5.1a: Test data.**

**Name: AE-4.0 DE & AE-2.5 DE** (4mm and 2.5mm fin pitch)

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 18.74359 | 38.57344 | 60.26727 | 56.29777 | 100304.5 | 2.3191244 | 2.766332329 |
| 2 | 19.5019 | 40.77964 | 60.57182 | 56.89674 | 100292.14 | 1.972282 | 2.746788075 |
| 3 | 19.40351 | 43.18701 | 60.50535 | 57.26594 | 100293.75 | 1.5331779 | 2.684653362 |
| 4 | 19.46064 | 46.38157 | 60.75912 | 57.99044 | 100292.81 | 1.1512463 | 2.696304122 |
| 5 | 19.31336 | 49.05874 | 60.79094 | 58.43243 | 100295.22 | 0.8866112 | 2.684422916 |
| 6 | 19.22826 | 52.31005 | 60.9337 | 59.07283 | 100296.6 | 0.6224236 | 2.675602114 |

**Table D.5.1b: Test data.**

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 700 | 378 | 1420 |
| 2 | 546 | 741 | 1082 |
| 3 | 359 | 448 | 687 |
| 4 | 226 | 254 | 414 |
| 5 | 148 | 152 | 262 |
| 6 | 85 | 76 | 142 |

**Table D.5.2: Pressure drops for isothermal tests.**

($p_{atm}$ =100700 N/m$^2$, $T_{aiwb}$ = 13.5 °C, $T_{aidb}$ = 16.5 °C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 733 | 416 | 1511 |
| 2 | 540 | 746 | 1084 |
| 3 | 346 | 443 | 668 |
| 4 | 216 | 250 | 402 |
| 5 | 140.5 | 149 | 252 |
| 6 | 82 | 77 | 140 |

**Table D.5.3a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 28.90232967 | 0.9941802 | 46727.283 | 45933.52 | 0.9830129 | 1.6987149 |
| 2 | 27.66644516 | 0.9939555 | 42632.059 | 42228.947 | 0.9905444 | 0.9455588 |
| 3 | 26.26472348 | 0.9941692 | 37046.673 | 36381.632 | 0.9820486 | 1.7951425 |
| 4 | 24.50104676 | 0.9946919 | 31489.478 | 31232.124 | 0.9918273 | 0.8172672 |
| 5 | 22.74141567 | 0.9950639 | 26798.497 | 26488.915 | 0.9884478 | 1.1552236 |
| 6 | 20.39946082 | 0.9949696 | 20925.794 | 20832.394 | 0.9955366 | 0.4463386 |

**Table D.5.3b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 113350.7401 | 11575.402 | 535120.3 | 318088.99 | 16.538449 |
| 2 | 113342.4858 | 11538.99 | 453343.8 | 300944.11 | 17.750011 |
| 3 | 111038.5288 | 11329.7 | 351420.45 | 273144.48 | 19.239769 |
| 4 | 112365.4392 | 11405.425 | 262825.39 | 246007.9 | 21.367383 |
| 5 | 112278.5191 | 11379.695 | 201793.98 | 223558.64 | 23.495152 |
| 6 | 112583.1554 | 11375.429 | 141124.41 | 192370.18 | 27.239267 |

**Correlating equations:** $Ny = 2268.266Ry^{0.37518}$

$K_{he} = 2103.001Ry^{-0.3673}$

**Table D.5.4: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 600304.7199 | 15.306581 |
| 2 | 490276.7922 | 16.905531 |
| 3 | 379223.3951 | 18.105211 |
| 4 | 285563.5814 | 19.932687 |
| 5 | 220545.8536 | 21.736833 |
| 6 | 158270.2086 | 24.633903 |

**Correlating equation:** $K_{heiso} = 1529.082Ry^{-0.3452}$

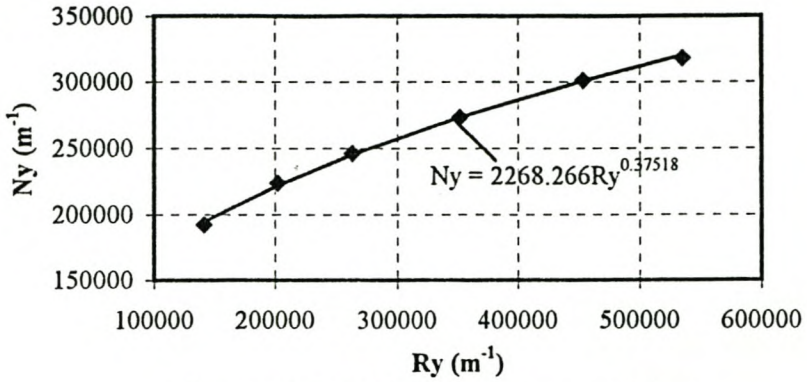



**Figure D.5.1: Heat transfer coefficient (4.0mm & 2.5mm pitch).**

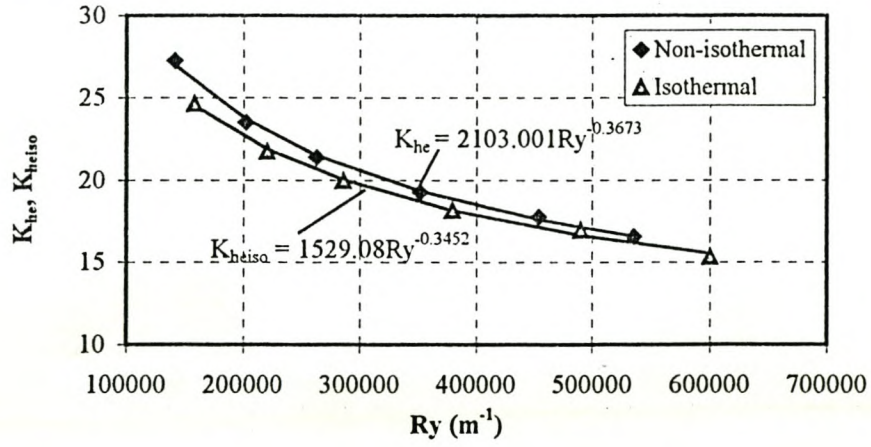



**Figure D.5.2: Pressure drop coefficient (4.0mm & 2.5mm pitch).**

**Table D.6.1a: Test data.**

**Name: AE-4.3 DA & AE-2.5 DE** (4.3mm and 2.5mm fin pitch)

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 20.23787 | 39.481 | 60.43598 | 56.42867 | 101446.31 | 2.3292644 | 2.696660247 |
| 2 | 20.58714 | 40.9979 | 60.45496 | 56.77465 | 101440.55 | 1.983066 | 2.676553685 |
| 3 | 20.54239 | 43.70613 | 60.64041 | 57.43307 | 101441.29 | 1.540773 | 2.65767884 |
| 4 | 20.35085 | 46.60748 | 60.81694 | 58.08058 | 101444.45 | 1.1574643 | 2.663203682 |
| 5 | 20.10528 | 49.11873 | 60.85023 | 58.50358 | 101448.5 | 0.89164 | 2.656653018 |
| 6 | 19.81653 | 52.24388 | 60.95499 | 59.08623 | 101453.26 | 0.6260712 | 2.641734513 |

**Table D.6.1b: Test data.**

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 706 | 380 | 1429 |
| 2 | 553 | 751 | 1098 |
| 3 | 357 | 445 | 680 |
| 4 | 223 | 250 | 406 |
| 5 | 146 | 148 | 255 |
| 6 | 84 | 74 | 140 |

**Table D.6.2: Pressure drops for isothermal tests.**

($p_{atm}$ =101880 N/m$^2$, $T_{aiwb}$ = 15 °C, $T_{aidb}$ = 17°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 745 | 415 | 1524 |
| 2 | 551 | 757 | 1120 |
| 3 | 357 | 456 | 690 |
| 4 | 213 | 246 | 393 |
| 5 | 146 | 154 | 260 |
| 6 | 88 | 82 | 148 |

**Table D.6.3a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 27.88254413 | 0.9938709 | 45656.924 | 45204.14 | 0.9900829 | 0.9917082 |
| 2 | 26.96267675 | 0.9938342 | 41390.851 | 41207.002 | 0.9955582 | 0.4441779 |
| 3 | 25.63050818 | 0.9940862 | 36141.048 | 35660.327 | 0.9866988 | 1.3301239 |
| 4 | 24.08530061 | 0.9946609 | 30638.252 | 30488.943 | 0.9951267 | 0.4873279 |
| 5 | 22.48966695 | 0.9950515 | 25941.101 | 26083.327 | 1.0054827 | -0.548265 |
| 6 | 20.29319566 | 0.995003 | 20360.501 | 20655.964 | 1.0145116 | -1.45116 |

**Table D.6.3b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 110753.7444 | 11340.259 | 537201.62 | 322045.85 | 16.572975 |
| 2 | 110239.9372 | 11281.936 | 458090.44 | 299729.49 | 17.713392 |
| 3 | 110179.9233 | 11244.423 | 351275.3 | 272595.67 | 19.236271 |
| 4 | 111112.433 | 11292.736 | 261820.2 | 243120.93 | 21.393373 |
| 5 | 111228.3201 | 11285.169 | 200046.94 | 218418.47 | 23.77469 |
| 6 | 111187.5731 | 11255.491 | 139984.52 | 187912.38 | 27.619274 |

**Correlating equations:** $Ny = 1740.4518Ry^{0.395564}$

$K_{he} = 2288.692Ry^{-0.3737}$

**Table D.6.4: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 602009.409 | 15.592907 |
| 2 | 495786.4628 | 17.003552 |
| 3 | 386240.6393 | 18.152216 |
| 4 | 284431.1905 | 19.971126 |
| 5 | 225130.2108 | 21.850565 |
| 6 | 164024.5017 | 24.810905 |

**Correlating equation:** $K_{heiso} = 1553.914Ry^{-0.34554}$

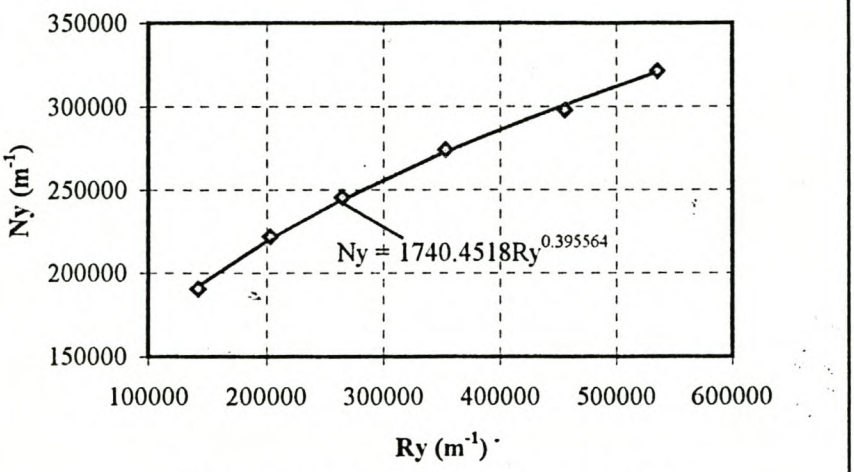



**Figure D.6.1: Heat transfer parameter (4.3mm & 2.5mm pitch).**

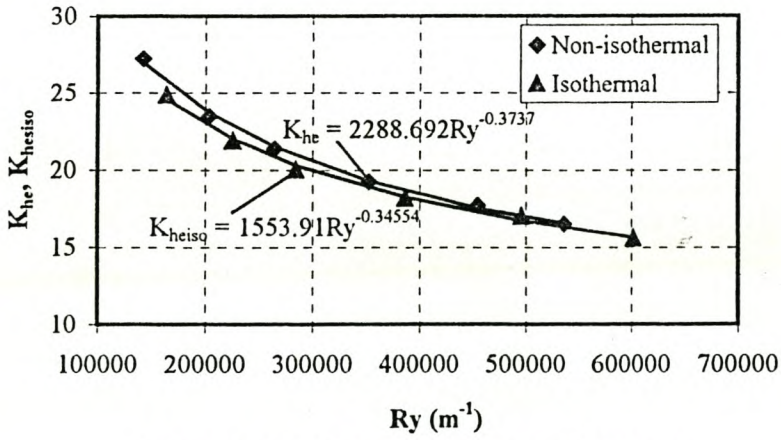



**Figure D.6.2: Pressure drop coefficient (4.3mm & 2.5mm pitch).**

**Table D.7.1a: Test data.**

**Name: AE-4.5 DA & AE-2.5 DE** (4.5mm and 2.5mm fin pitch)

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 20.4924 | 39.62135 | 60.41722 | 56.58973 | 101073.34 | 2.3317917 | 2.736249786 |
| 2 | 20.82835 | 41.3171 | 60.49746 | 56.98384 | 101067.82 | 1.9899757 | 2.716681336 |
| 3 | 21.21732 | 44.11418 | 60.72111 | 57.65881 | 101061.43 | 1.5479413 | 2.717167134 |
| 4 | 21.17278 | 46.97646 | 60.74664 | 58.16289 | 101062.16 | 1.1507769 | 2.694865551 |
| 5 | 20.8809 | 49.23565 | 60.88678 | 58.61902 | 101066.96 | 0.9048953 | 2.667432046 |
| 6 | 20.72247 | 52.28189 | 60.92827 | 59.11453 | 101069.56 | 0.6456072 | 2.692434005 |

**Table D.7.1b: Test data.**

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 692 | 380 | 1414 |
| 2 | 539 | 749 | 1080 |
| 3 | 357 | 454 | 690 |
| 4 | 219 | 252 | 403 |
| 5 | 150 | 157 | 266 |
| 6 | 87.5 | 81 | 147 |

**Table D.7.2: Pressure drops for isothermal tests.**

($p_{atm}$ = 101390 N/m$^2$, $T_{aiwb}$ = 17 °C, $T_{aidb}$ = 20 °C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 728 | 409 | 1507 |
| 2 | 525 | 745 | 1058 |
| 3 | 346 | 457 | 674 |
| 4 | 208 | 246 | 387 |
| 5 | 141 | 152 | 254 |
| 6 | 90 | 87 | 154 |

**Table D.7.3a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 27.74695419 | 0.9941304 | 45249.803 | 43810.011 | 0.9681813 | 3.1818728 |
| 2 | 26.7771041 | 0.9940429 | 41358.841 | 39931.186 | 0.9654812 | 3.4518751 |
| 3 | 25.23840924 | 0.9942618 | 35950.851 | 34810.629 | 0.9682839 | 3.171613 |
| 4 | 23.49850014 | 0.9948346 | 30122.316 | 29130.822 | 0.9670844 | 3.2915611 |
| 5 | 22.19663934 | 0.9951676 | 26031.901 | 25309.02 | 0.972231 | 2.7769013 |
| 6 | 19.95403625 | 0.9951235 | 20674.535 | 20432.567 | 0.9882963 | 1.1703661 |

**Table D.7.3b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 112504.0367 | 11485.24 | 535595.38 | 319942.71 | 16.239807 |
| 2 | 112111.238 | 11432.971 | 455905.35 | 301021.98 | 17.310386 |
| 3 | 112912.8699 | 11466.526 | 353221.03 | 274645.86 | 18.8503 |
| 4 | 112443.794 | 11405.914 | 261695.29 | 244438.12 | 20.826828 |
| 5 | 111809.8782 | 11328.91 | 205302.98 | 221869.33 | 22.996265 |
| 6 | 113322.8254 | 11436.803 | 145970.46 | 193872.78 | 26.23002 |

**Correlating equations:** $Ny = 1995.571Ry^{0.385128}$

$K_{he} = 1989.517Ry^{-0.36458}$

**Table D.7.4: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 588670.0634 | 15.457794 |
| 2 | 484579.7327 | 16.450869 |
| 3 | 380847.916 | 17.552256 |
| 4 | 280143.8445 | 19.501193 |
| 5 | 220266.5509 | 21.38366 |
| 6 | 166413.8492 | 23.912424 |

**Correlating equation:** $K_{heiso} = 1452.0854Ry^{-0.34264}$

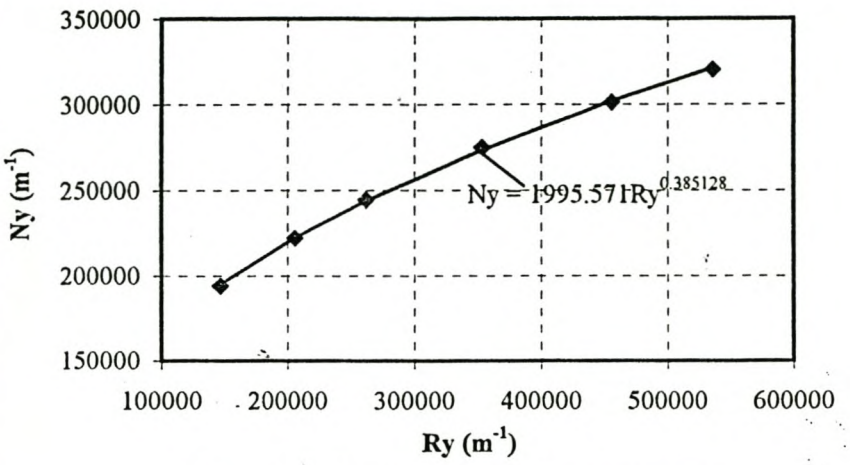



**Figure D.7.1: Heat transfer parameter (4.5mm & 2.5mm pitch).**

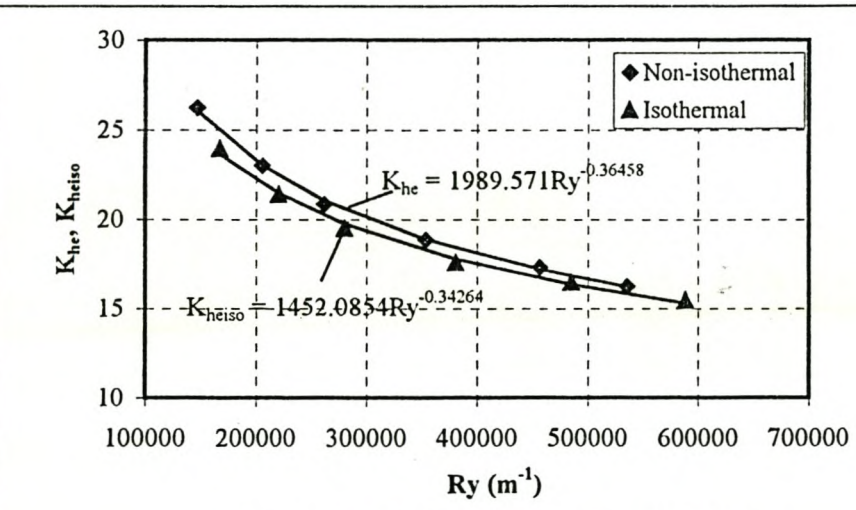



**Figure D.7.2: Pressure drop coefficient (4.5mm & 2.5mm pitch).**

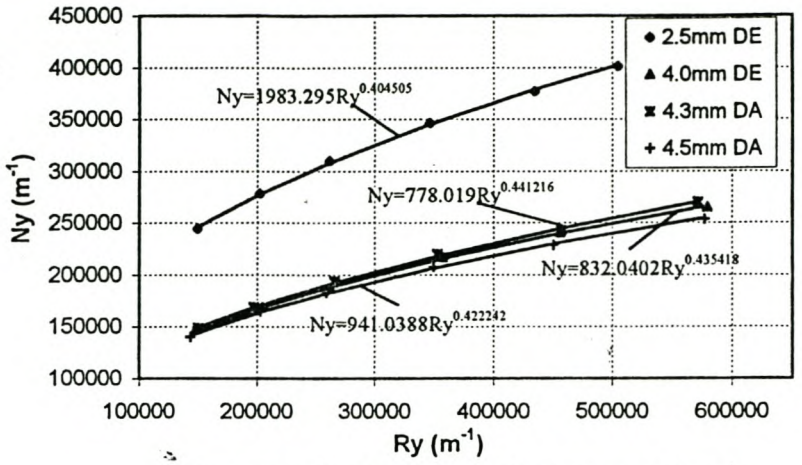



**Figure D.8: Heat transfer parameter (Double-Row).**

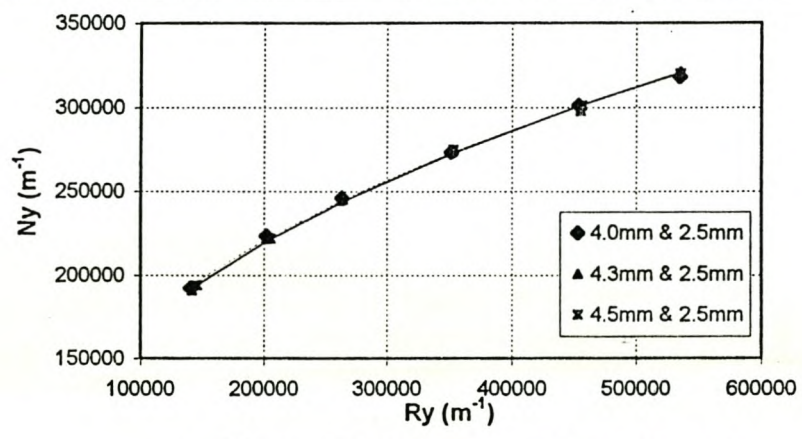



**Figure D.9: Heat transfer parameter (Combination).**

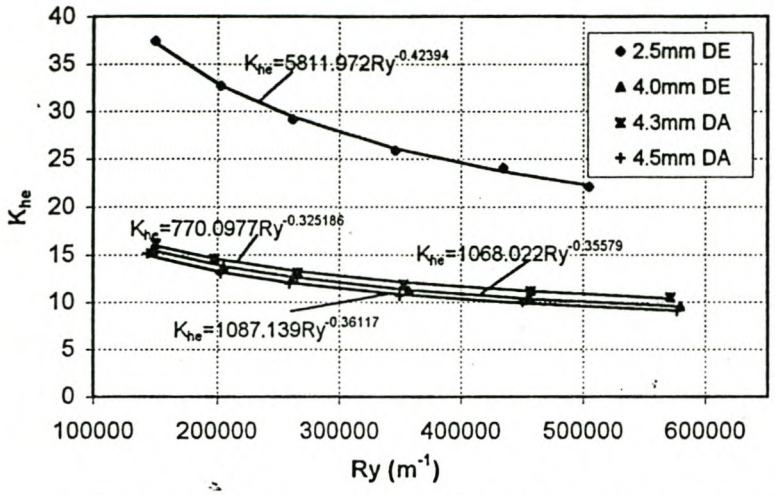



**Figure D.10: Pressure drop coefficient (Double row).**

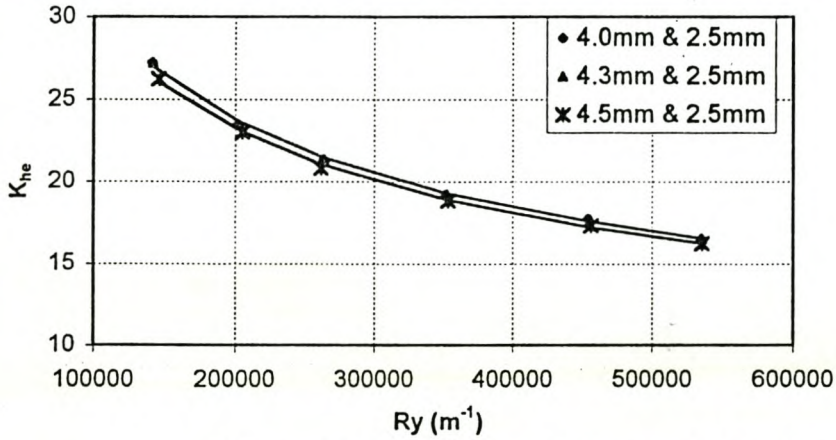



**Figure D.11: Pressure drop coefficient (Combination).**

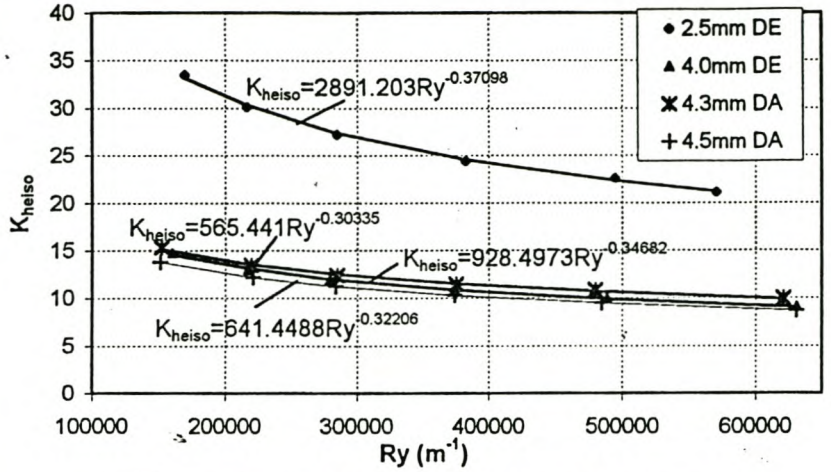



**Figure D.12: Isothermal pressure drop coefficient (Double row).**

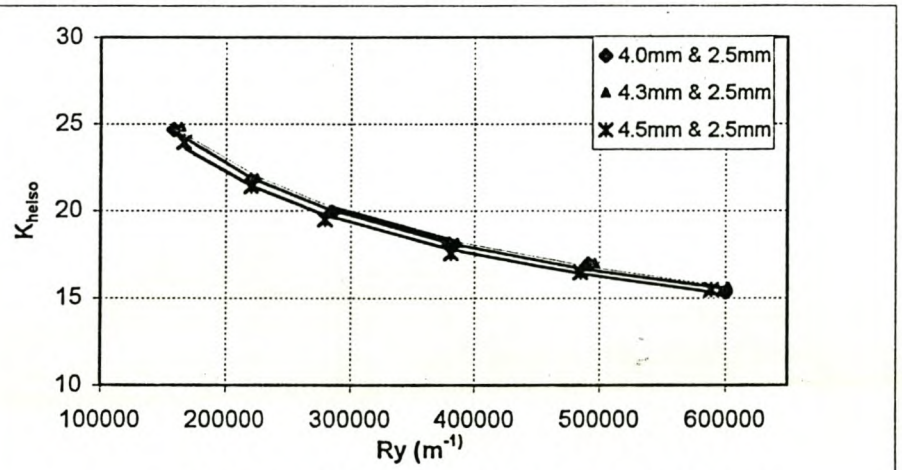



**Figure D.13: Isothermal Pressure drop coefficient (Combination).**

# APPENDIX E

## Thermal-flow performance of ED-, F- and R-finned tubes

## Table E.1.1: Tube specifications.

Name: ED-2.5L (2.5 mm fin pitch, 0.61846 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 2.19 | 200 | 495 | 2.47500 |
| 2 | 2.19 | 199 | 500 | 2.51256 |
| 3 | 2.19 | 199 | 496 | 2.49246 |
| 4 | 2.19 | 198 | 496 | 2.50505 |
| 5 | 2.195 | 200 | 496 | 2.48000 |
| 6 | 2.195 | 200 | 497 | 2.48500 |
| 7 | 2.18 | 199 | 497 | 2.49749 |
| 8 | 2.18 | 200 | 499 | 2.49500 |
| 9 | 2.185 | 199 | 494 | 2.48241 |
| 10 | 2.205 | 200 | 498 | 2.49000 |
| Average | 2.19 | 199.4 | 496.8 | 2.4914975 |

## Table E.1.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 26.98155 | 44.68019 | 59.830495 | 56.232685 | 99921.325 | 2.2715039 | 2.6829761 |
| 2 | 27.33121 | 46.49938 | 60.158635 | 56.853585 | 99915.64 | 1.9358056 | 2.6558297 |
| 3 | 27.378 | 48.59884 | 60.205515 | 57.257475 | 99914.88 | 1.5060331 | 2.627708 |
| 4 | 27.47149 | 50.69472 | 60.172075 | 57.673035 | 99913.36 | 1.1609276 | 2.6754144 |
| 5 | 27.46811 | 52.92584 | 60.385805 | 58.272835 | 99913.415 | 0.892454 | 2.660742 |
| 6 | 27.63577 | 55.31053 | 60.404725 | 58.746605 | 99910.689 | 0.6256489 | 2.6477528 |

## Table E.1.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 570 | 373 | 1718 |
| 2 | 528 | 736 | 1750 |
| 3 | 368 | 444 | 1096 |
| 4 | 223 | 264 | 680 |
| 5 | 152 | 157 | 420 |
| 6 | 92 | 78 | 224 |

## Table E.1.3: Pressure drops for isothermal tests.

($p_{atm}$ = 100300 N/m$^2$, $T_{aiwb}$ = 20°C, $T_{aidb}$ = 24°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 426 | 400 | 1200 |
| 2 | 320 | 756 | 890 |
| 3 | 210 | 454 | 543 |
| 4 | 131 | 250 | 312 |
| 5 | 89 | 154 | 215 |
| 6 | 56 | 82 | 110 |

**Table E.1.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 21.43316466 | 0.9985728 | 40969.643 | 40376.516 | 0.9855228 | 1.4477221 |
| 2 | 20.58191144 | 0.998611 | 37811.557 | 36718.251 | 0.9710854 | 2.8914611 |
| 3 | 19.32420229 | 0.9983243 | 32568.606 | 32406.237 | 0.9950146 | 0.4985422 |
| 4 | 17.88128379 | 0.9978654 | 27475.381 | 27970.143 | 1.0180075 | -1.800748 |
| 5 | 16.46179687 | 0.9973123 | 23155.345 | 23520.849 | 1.0157849 | -1.578488 |
| 6 | 14.37816761 | 0.9968155 | 17646.912 | 18368.196 | 1.0408731 | -4.087309 |

**Table E.1.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 195468.7295 | 23449.242 | 515888.77 | 352421.03 | 13.634871 |
| 2 | 194926.0282 | 23315.139 | 438461.71 | 337292.15 | 17.330229 |
| 3 | 193537.6314 | 23137.588 | 340234.51 | 307835.49 | 19.887208 |
| 4 | 197635.0543 | 23518.915 | 261577.24 | 279054.45 | 20.210057 |
| 5 | 197788.9969 | 23466.541 | 200551.41 | 254250.96 | 23.227271 |
| 6 | 197570.6283 | 23403.782 | 140163.67 | 220494.11 | 28.490071 |

**Correlating equations:** $Ny = 3039.7200Ry^{0.36213}$

**Table E.1.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 571362.0432 | 13.759583 |
| 2 | 477196.1601 | 14.995303 |
| 3 | 380012.4885 | 16.576413 |
| 4 | 285565.1238 | 19.253155 |
| 5 | 209515.3955 | 23.256454 |
| 6 | 154972.8439 | 28.43596 |

**Correlating equation:** $K_{heiso} = 13216.347Ry^{-0.518865}$

**Table E.2.1: Tube specifications.**

Name: ED-2.5S (2.5 mm fin pitch, 0.53111 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 1.105 | 188 | 468 | 2.48936 |
| 2 | 1.145 | 188 | 469 | 2.49468 |
| 3 | 1.19 | 189 | 467 | 2.47090 |
| 4 | 1.105 | 189 | 467 | 2.47090 |
| 5 | 1.195 | 188 | 468 | 2.48936 |
| 6 | 1.19 | 188 | 468 | 2.48936 |
| 7 | 1.105 | 192 | 468 | 2.43750 |
| 8 | 1.2 | 189 | 469 | 2.48148 |
| 9 | 1.2 | 191 | 469 | 2.45550 |
| 10 | 1.2 | 189 | 469 | 2.48148 |
| Average | 1.1635 | 189.1 | 468.2 | 2.4760525 |

**Table E.2.2a: Test data.**

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 27.72707 | 44.24221 | 59.99608 | 56.50789 | 100028.67 | 2.3375712 | 2.6844607 |
| 2 | 28.12587 | 45.87716 | 60.14224 | 57.00182 | 100022.17 | 1.9357609 | 2.6478809 |
| 3 | 28.40576 | 47.52677 | 60.05287 | 57.28438 | 100017.62 | 1.5398504 | 2.6361929 |
| 4 | 28.25957 | 49.451 | 60.27647 | 57.89052 | 100020 | 1.1837585 | 2.6594341 |
| 5 | 28.12684 | 51.4501 | 60.41146 | 58.43543 | 100022.16 | 0.8866234 | 2.6332658 |
| 6 | 28.25523 | 54.35312 | 60.51618 | 59.09616 | 100020.07 | 0.5676216 | 2.6368504 |

**Table E.2.2b: Test data.**

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 500 | 394 | 1714 |
| 2 | 460 | 733 | 1676 |
| 3 | 330 | 462 | 1074 |
| 4 | 253 | 273 | 654 |
| 5 | 163 | 154 | 380 |
| 6 | 102 | 64 | 167 |

**Table E.2.3: Pressure drops for isothermal tests.**

($p_{atm}$ = 100500 N/m$^2$, $T_{aiwb}$ = 20°C, $T_{aidb}$ = 25°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 512 | 410 | 1796 |
| 2 | 483 | 764 | 1734 |
| 3 | 325 | 476 | 1106 |
| 4 | 240 | 264 | 635 |
| 5 | 174 | 152 | 375 |
| 6 | 122 | 92 | 240 |

**Table E.2.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 21.61707551 | 0.9986563 | 39331.324 | 39169.21 | 0.9958783 | 0.4121732 |
| 2 | 20.71892293 | 0.9988395 | 35005.574 | 34785.168 | 0.9937037 | 0.6296325 |
| 3 | 19.57712594 | 0.9988104 | 29993.441 | 30530.523 | 1.0179066 | -1.790664 |
| 4 | 18.67628248 | 0.998583 | 25556.736 | 26545.532 | 1.0386902 | -3.869023 |
| 5 | 17.51913776 | 0.9982348 | 21069.54 | 21769.643 | 1.0332282 | -3.322824 |
| 6 | 15.32530618 | 0.9977846 | 15094.247 | 15666.37 | 1.0379034 | -3.790337 |

**Table E.2.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 280375.9704 | 45883.178 | 530598.45 | 330351.5 | 11.302577 |
| 2 | 277935.0898 | 45437.739 | 438274.92 | 305472.78 | 15.113997 |
| 3 | 277123.2708 | 45293.231 | 347806.19 | 275815.87 | 17.082161 |
| 4 | 281366.9275 | 45742.466 | 266815.29 | 245341 | 22.097131 |
| 5 | 280061.9319 | 45450.114 | 199402.81 | 214790.28 | 25.30167 |
| 6 | 282096.1656 | 45603.21 | 127195.27 | 174844.22 | 38.443986 |

**Correlating equations:** $Ny = 919.5229Ry^{0.446805}$

**Table E.2.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 573686.3178 | 10.878471 |
| 2 | 476657.2052 | 14.865574 |
| 3 | 378036.3222 | 15.902419 |
| 4 | 282536.361 | 21.023714 |
| 5 | 214564.3464 | 26.429018 |
| 6 | 166756.8703 | 30.678862 |

**Correlating equation:** $K_{heiso} = 867074.55Ry^{-0.846683}$

## Table E.3.1: Tube specifications.

Name: F-3.0 (3.0 mm fin pitch, 0.30609 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 0.955 | 157 | 467 | 2.975 |
| 2 | 0.965 | 158 | 466 | 2.949 |
| 3 | 0.98 | 158 | 468 | 2.962 |
| 4 | 0.96 | 158 | 468 | 2.962 |
| 5 | 0.97 | 158 | 468 | 2.962 |
| 6 | 0.96 | 158 | 467 | 2.956 |
| 7 | 0.965 | 158 | 468 | 2.962 |
| 8 | 0.965 | 158 | 468 | 2.962 |
| 9 | 0.95 | 158 | 467 | 2.956 |
| 10 | 0.965 | 158 | 468 | 2.962 |
| Average | 0.9635 | 157.9 | 467.5 | 2.9607434 |

## Table E.3.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 25.55694 | 40.85775 | 60.1856 | 57.00498 | 100004.24 | 2.2711979 | 2.7168352 |
| 2 | 25.76049 | 42.13849 | 60.26346 | 57.29422 | 100000.93 | 1.9487529 | 2.7169084 |
| 3 | 25.76083 | 44.01583 | 60.2806 | 57.62017 | 100000.92 | 1.5478272 | 2.7059731 |
| 4 | 24.8886 | 45.96716 | 60.28356 | 57.98824 | 100015.11 | 1.1769354 | 2.6782224 |
| 5 | 25.3336 | 48.62603 | 60.43309 | 58.50058 | 100007.87 | 0.8788067 | 2.7128158 |
| 6 | 25.25841 | 51.74997 | 60.57148 | 59.08844 | 100009.1 | 0.5962203 | 2.7048417 |

## Table E.3.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 628 | 368 | 1745 |
| 2 | 496 | 735 | 1786 |
| 3 | 366 | 462 | 1140 |
| 4 | 250 | 267 | 675 |
| 5 | 162 | 150 | 392 |
| 6 | 103 | 70 | 186 |

## Table E.3.3: Pressure drops for isothermal tests.

($p_{atm}$ = 100200 N/m$^2$, $T_{aiwb}$ = 24°C, $T_{aidb}$ = 27°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 645 | 378 | 1805 |
| 2 | 538 | 737 | 1776 |
| 3 | 358 | 437 | 1075 |
| 4 | 242 | 260 | 669 |
| 5 | 194 | 179 | 462 |
| 6 | 120 | 78 | 208 |

**Table E.3.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 24.89821069 | 0.9994333 | 35384.988 | 36148.019 | 1.0215637 | -2.15637 |
| 2 | 24.21371199 | 1.0004546 | 32497.758 | 33747.49 | 1.0384559 | -3.845592 |
| 3 | 23.19484369 | 1.0014246 | 28771.474 | 30116.71 | 1.0467559 | -4.675592 |
| 4 | 22.41124727 | 1.0013194 | 25269.552 | 25717.819 | 1.0177394 | -1.773942 |
| 5 | 20.68042043 | 1.00011 | 20848.846 | 21933.503 | 1.0520248 | -5.20248 |
| 6 | 18.60537281 | 0.9980167 | 16089.331 | 16783.585 | 1.0431499 | -4.314995 |

**Table E.3.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 337758.0576 | 51568.709 | 540504.27 | 270185.41 | 13.985068 |
| 2 | 338729.7489 | 51624.803 | 462907.84 | 254103.73 | 14.967159 |
| 3 | 338263.2036 | 51499.189 | 366833.51 | 233720.05 | 17.453467 |
| 4 | 335754.9702 | 51104.648 | 278624.28 | 211889.02 | 20.582098 |
| 5 | 341830.0607 | 51763.263 | 207246.07 | 188556.85 | 23.802363 |
| 6 | 342730.3425 | 51741.018 | 140092.88 | 161122.98 | 32.71775 |

**Correlating equations:** $Ny = 1794.6929Ry^{0.379978}$

**Table E.3.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 568907.1506 | 13.696957 |
| 2 | 483424.3457 | 15.822402 |
| 3 | 374199.4956 | 17.572106 |
| 4 | 289513.1571 | 19.84385 |
| 5 | 240427.1901 | 23.066502 |
| 6 | 158474.9029 | 32.840324 |

**Correlating equation:** $K_{heiso} = 24057.539Ry^{-0.562371}$

## Table E.4.1: Tube specifications.

**Name: R-2.4G** (2.4 mm fin pitch, 0.1415 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 1.62 | 287 | 700 | 2.439 |
| 2 | 1.61 | 290 | 702 | 2.421 |
| 3 | 1.62 | 274 | 697 | 2.544 |
| 4 | 1.605 | 287 | 700 | 2.439 |
| 5 | 1.61 | 287 | 702 | 2.446 |
| 6 | 1.61 | 287 | 700 | 2.439 |
| 7 | 1.615 | 290 | 702 | 2.421 |
| 8 | 1.61 | 287 | 700 | 2.439 |
| 9 | 1.605 | 287 | 700 | 2.439 |
| 10 | 1.615 | 284 | 699 | 2.461 |
| Average | 1.612 | 286 | 700.2 | 2.4487558 |

## Table E.4.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 28.7483 | 35.78348 | 59.28378 | 57.25881 | 99902.554 | 2.4907263 | 2.160680298 |
| 2 | 28.88517 | 36.79305 | 59.40965 | 57.60362 | 99900.329 | 2.004592 | 2.137262393 |
| 3 | 27.98609 | 37.303 | 59.44003 | 57.79891 | 99914.948 | 1.5531638 | 2.124104739 |
| 4 | 27.61519 | 39.06605 | 59.42158 | 57.99561 | 99920.979 | 1.1471432 | 2.240442187 |
| 5 | 27.51316 | 41.75605 | 59.49513 | 58.11019 | 99922.638 | 0.9056368 | 2.231560957 |
| 6 | 27.76831 | 44.07007 | 59.51234 | 58.38744 | 99918.489 | 0.6241827 | 2.18077852 |

## Table E.4.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 424 | 435 | 1278 |
| 2 | 298 | 760 | 857 |
| 3 | 195 | 454 | 528 |
| 4 | 120 | 248 | 298 |
| 5 | 82 | 156 | 194 |
| 6 | 46 | 75 | 97 |

## Table E.4.3: Pressure drops for isothermal tests.

($p_{atm}$ = 1005700 N/m$^2$, $T_{aiwb}$ = 21.5°C, $T_{aidb}$ = 25°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 430 | 452 | 1324 |
| 2 | 288 | 757 | 848 |
| 3 | 192 | 460 | 532 |
| 4 | 116 | 247 | 296 |
| 5 | 75 | 144 | 178 |
| 6 | 47 | 81 | 104 |

**Table E.4.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 25.92476592 | 0.9989907 | 17898.622 | 18301.975 | 1.0225354 | -2.253543 |
| 2 | 25.54618523 | 0.998534 | 16191.886 | 16146.804 | 0.9972158 | 0.2784242 |
| 3 | 25.78479057 | 0.9986576 | 14785.385 | 14582.34 | 0.9862672 | 1.3732792 |
| 4 | 25.03433034 | 0.9991921 | 13423.715 | 13364.748 | 0.9956073 | 0.4392707 |
| 5 | 23.58682011 | 0.9996535 | 13183.085 | 12928.925 | 0.9807207 | 1.9279287 |
| 6 | 22.17165213 | 0.9998447 | 10399.324 | 10262.606 | 0.9868533 | 1.3146747 |

**Table E.4.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 295862.5454 | 35462.368 | 595380.18 | 132358.04 | 7.8497763 |
| 2 | 293729.0519 | 35184.415 | 478487.94 | 121166.03 | 8.5015515 |
| 3 | 292432.5673 | 35023.622 | 370988.18 | 109456.44 | 9.2721124 |
| 4 | 308875.7493 | 36664.59 | 273564.02 | 101895.36 | 10.43576 |
| 5 | 308100.0382 | 36561.156 | 215296.29 | 105865.63 | 11.393327 |
| 6 | 301775.4934 | 35884.286 | 147916.02 | 88280.158 | 13.399214 |

**Correlating equations:** $Ny = 4082.773Ry^{0.259598}$

**Table E.4.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 629393.6605 | 7.6043941 |
| 2 | 496953.2905 | 8.1696271 |
| 3 | 388629.4006 | 8.9057564 |
| 4 | 285443.4092 | 9.9737583 |
| 5 | 217929.1651 | 11.062963 |
| 6 | 163167.9822 | 12.367128 |

**Correlating equation:** $K_{heiso} = 957.70213Ry^{-0.362919}$

## Table E.5.1: Tube specifications.

**Name: R-2.3D** (2.3 mm fin pitch, 0.2734 mm fin thickness)

| No | Tube mass,kg | No. of fins | $L_t$,mm | $P_f$,mm |
|---|---|---|---|---|
| 1 | 1.695 | 295 | .691 | 2.342 |
| 2 | 1.715 | 309 | 692 | 2.239 |
| 3 | 1.68 | 294 | 693 | 2.357 |
| 4 | 1.695 | 294 | 694 | 2.361 |
| 5 | 1.695 | 296 | 692 | 2.338 |
| 6 | 1.665 | 288 | 680 | 2.361 |
| 7 | 1.715 | 302 | 695 | 2.301 |
| 8 | 1.695 | 302 | 693 | 2.295 |
| 9 | 1.68 | 301 | 694 | 2.306 |
| 10 | 1.695 | 294 | 693 | 2.357 |
| Average | 1.693 | 297.5 | 691.7 | 2.3257308 |

## Table E.5.2a: Test data.

| Run no. | $T_{ai}$ °C | $T_{ao}$ °C | $T_{wi}$ °C | $T_{wo}$ °C | $p_{atm}$ N/m$^2$ | $m_a$ kg/s | $m_w$ kg/s |
|---|---|---|---|---|---|---|---|
| 1 | 27.05794 | 39.80523 | 59.21234 | 55.83373 | 99959.908 | 2.3585657 | 2.187686292 |
| 2 | 27.53387 | 41.32444 | 59.24683 | 56.27968 | 99952.167 | 1.9751743 | 2.176405094 |
| 3 | 27.76648 | 43.0028 | 59.29804 | 56.66443 | 99948.384 | 1.5510505 | 2.175058571 |
| 4 | 27.55348 | 45.29413 | 59.47701 | 57.27427 | 99951.848 | 1.1235785 | 2.175296548 |
| 5 | 27.59625 | 46.99948 | 59.50955 | 57.67888 | 99951.153 | 0.8993996 | 2.169949316 |
| 6 | 27.43821 | 49.92303 | 59.63024 | 58.15447 | 99953.723 | 0.6067256 | 2.157281007 |

## Table E.5.2b: Test data.

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 560 | 394 | 1340 |
| 2 | 416 | 747 | 969 |
| 3 | 274 | 460 | 615 |
| 4 | 159 | 242 | 335 |
| 5 | 110 | 156 | 222 |
| 6 | 61 | 72 | 108 |

## Table E.5.3: Pressure drops for isothermal tests.

($p_{atm}$ = 100350 N/m$^2$, $T_{aiwb}$ = 18.5°C, $T_{aidb}$ = 23°C)

| Run no. | $\Delta p_{bundle}$ N/m$^2$ | $\Delta p_n$ N/m$^2$ | $\Delta p_{up}$ N/m$^2$ |
|---|---|---|---|
| 1 | 582 | 435 | 1400 |
| 2 | 417 | 766 | 983 |
| 3 | 266 | 455 | 603 |
| 4 | 160 | 254 | 348 |
| 5 | 100 | 148 | 209 |
| 6 | 57 | 76 | 112 |

**Table E.5.4a: Test results.**

| Run no | LMTD °C | $F_T$ --- | $Q_a$ W | $Q_w$ W | $Q_w/Q_a$ --- | %error % |
|---|---|---|---|---|---|---|
| 1 | 23.78471934 | 0.9984551 | 30597.988 | 30914.732 | 1.0103518 | -1.035179 |
| 2 | 22.90955762 | 0.9988713 | 27718.202 | 27010.734 | 0.9744764 | 2.5523604 |
| 3 | 21.99818887 | 0.9992333 | 24047.52 | 23960.346 | 0.9963749 | 0.3625068 |
| 4 | 21.00253423 | 0.9994488 | 20285.916 | 20043.666 | 0.9880582 | 1.1941765 |
| 5 | 20.02767203 | 0.999414 | 17760.824 | 16617.627 | 0.9356338 | 6.4366225 |
| 6 | 18.2382377 | 0.9989881 | 13885.987 | 13318.456 | 0.9591292 | 4.0870788 |

**Table E.5.4b: Test results.**

| Run no. | $Re_w$ --- | $h_w$ $W/m^2 K$ | Ry $m^{-1}$ | Ny $m^{-1}$ | $K_{he}$ |
|---|---|---|---|---|---|
| 1 | 277483.8572 | 31710.739 | 560794.31 | 251315.43 | 11.554594 |
| 2 | 277094.3462 | 31617.171 | 468444.51 | 234988.95 | 12.19975 |
| 3 | 277868.4181 | 31641.529 | 366984.95 | 210791.76 | 12.990573 |
| 4 | 279612.4741 | 31718.329 | 265189.19 | 184830.04 | 14.317003 |
| 5 | 279873.9826 | 31692.881 | 211829.18 | 168835.51 | 15.414385 |
| 6 | 279528.4313 | 31590.973 | 142429.33 | 143851.11 | 18.700295 |

**Correlating equations:** $Ny = 1116.6703Ry^{0.409269}$

**Table E.5.5: Isothermal test results.**

| Run no. | Ry $m^{-1}$ | $K_{heiso}$ |
|---|---|---|
| 1 | 621343.2002 | 10.701522 |
| 2 | 502832.2553 | 11.707804 |
| 3 | 388939.0396 | 12.482576 |
| 4 | 291293.3256 | 13.385807 |
| 5 | 222390.0098 | 14.35341 |
| 6 | 159059.3029 | 15.993457 |

**Correlating equation:** $K_{heiso} = 430.2697Ry^{-0.275697}$

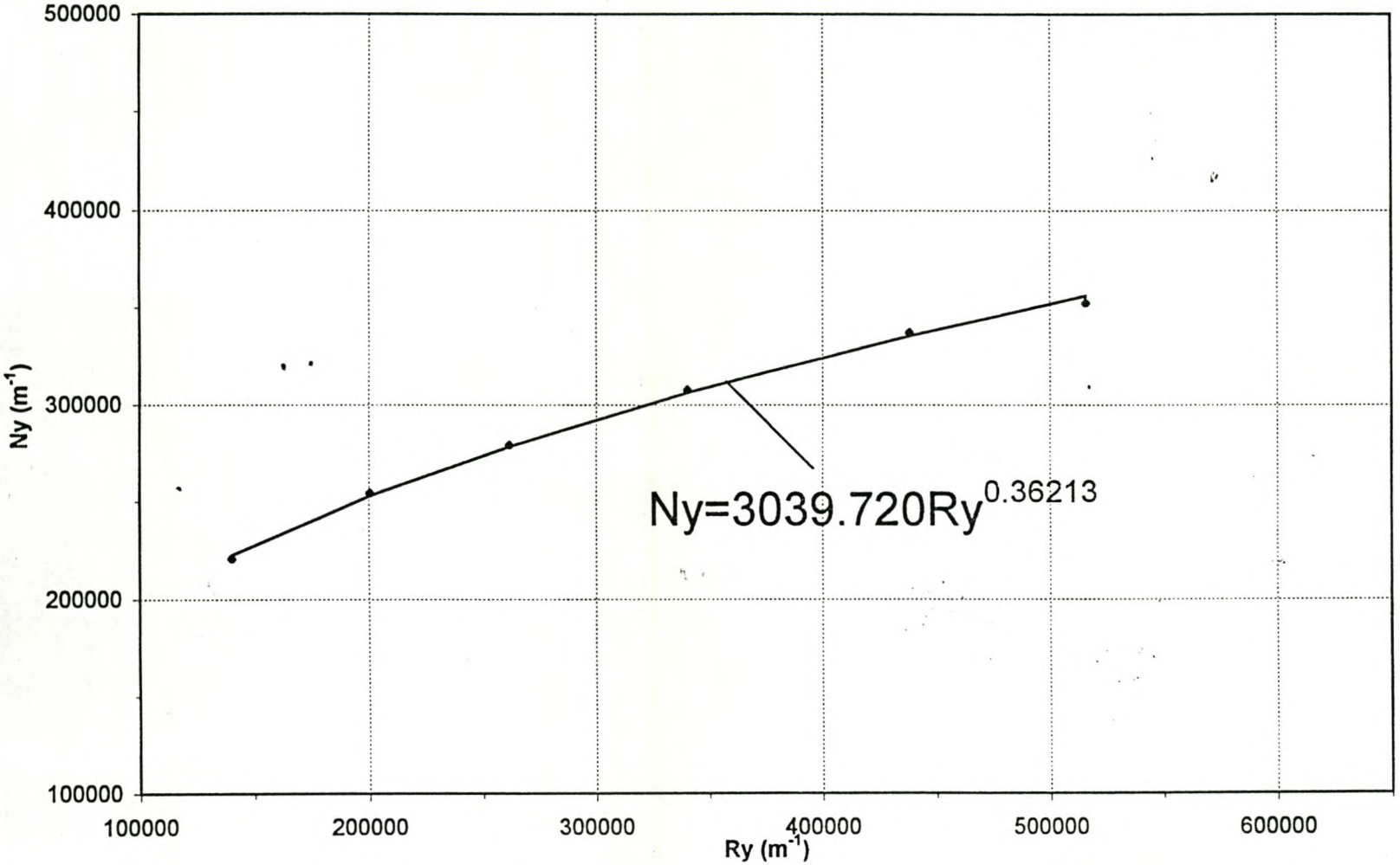



**Figure E.1: Heat transfer parameter (ED-2.5L).**

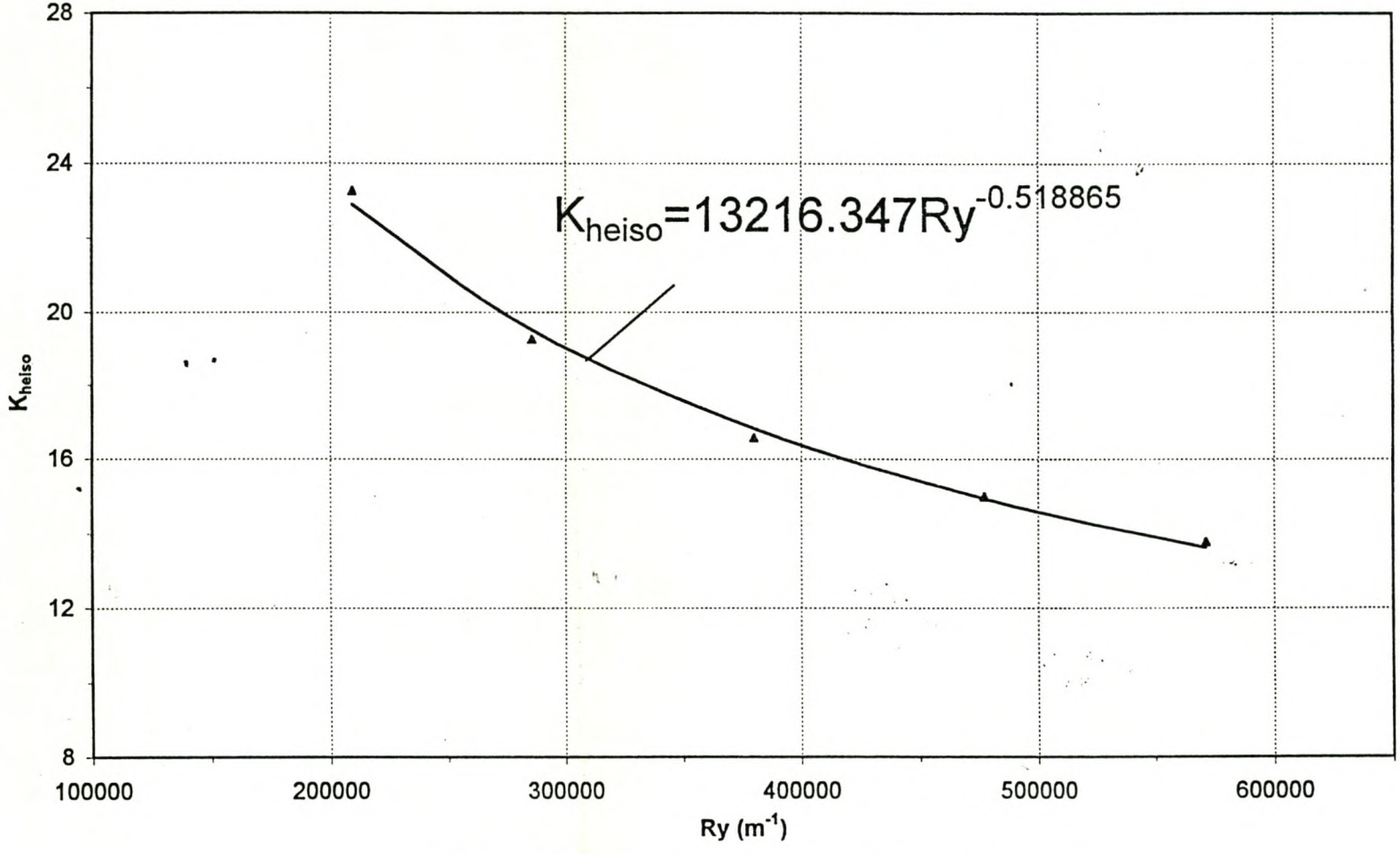



**Figure E.2: Pressure drop coefficient (ED-2.5L).**

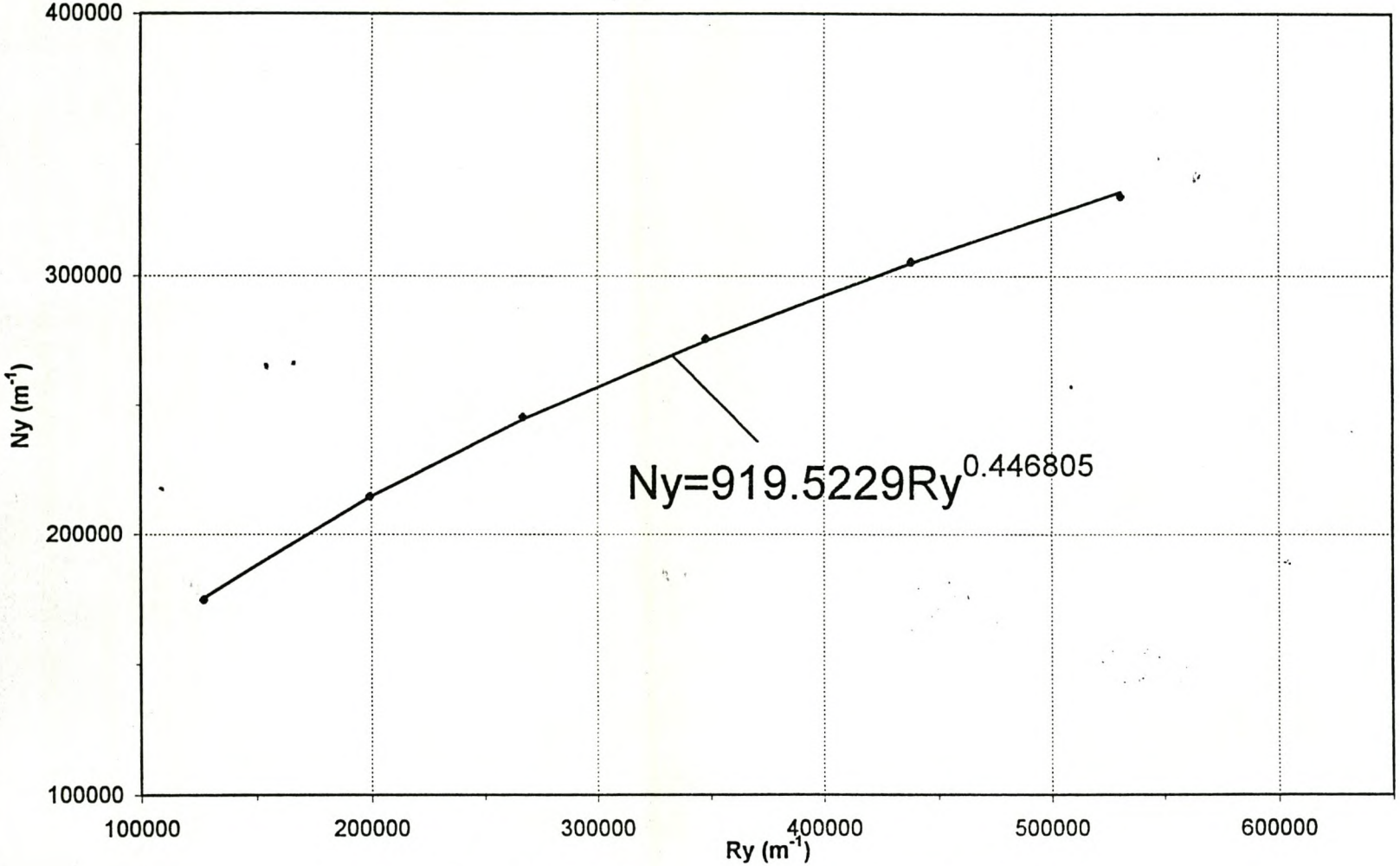



**Figure E.3: Heat transfer parameter (ED-2.5S).**

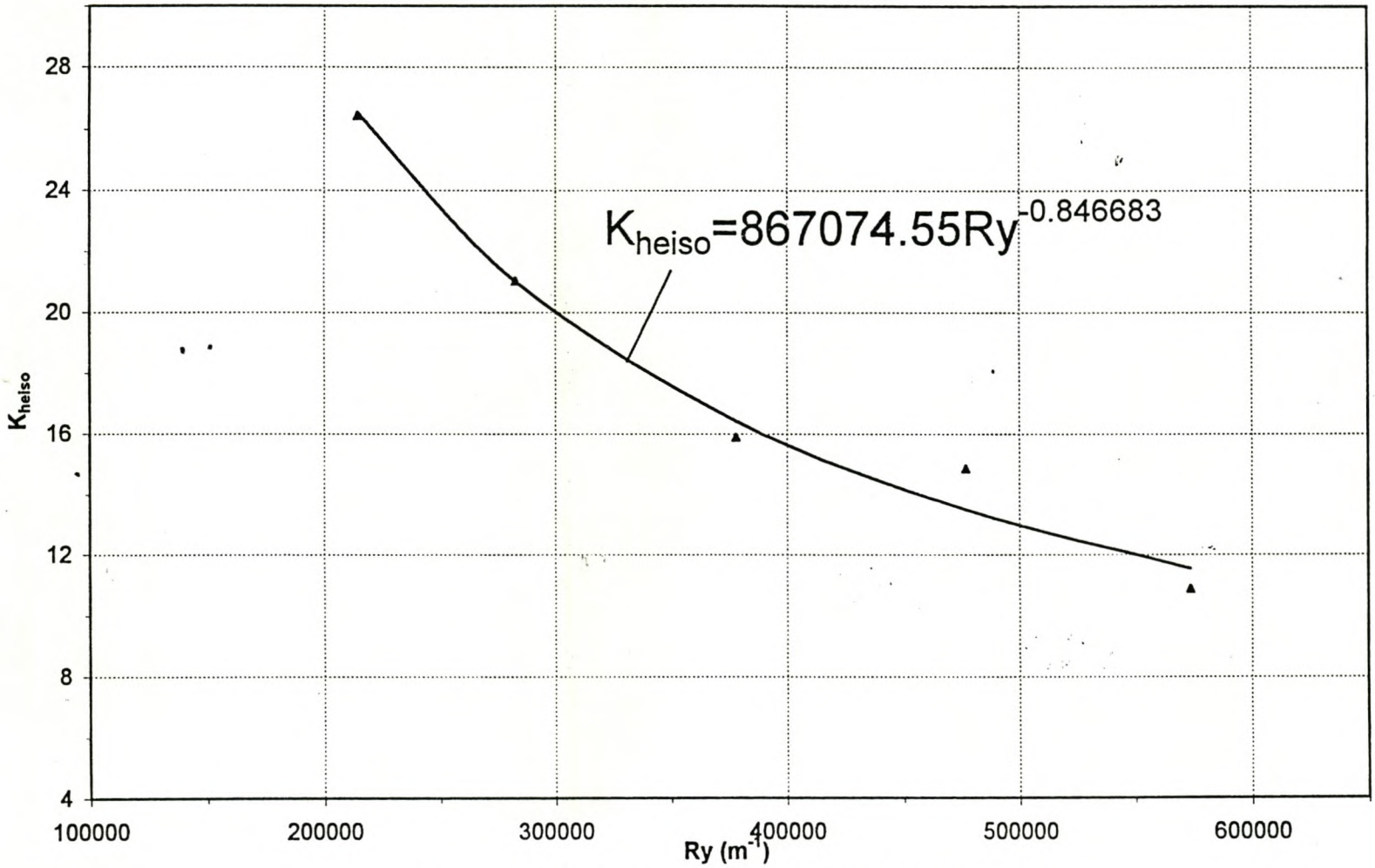



**Figure E.4: Pressure drop coefficient (ED-2.5S).**

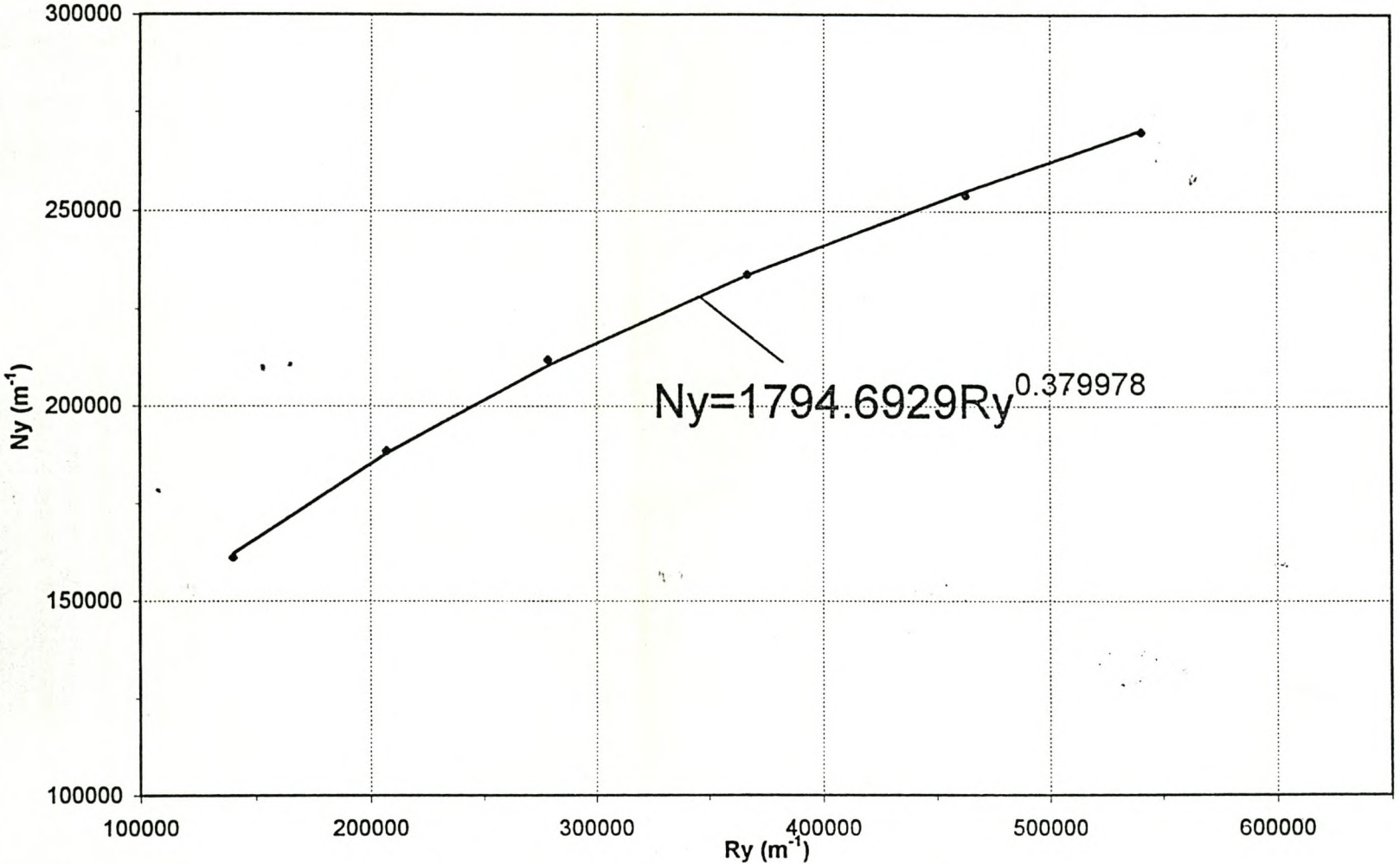



**Figure E.5: Heat transfer parameter (F- 3.0).**

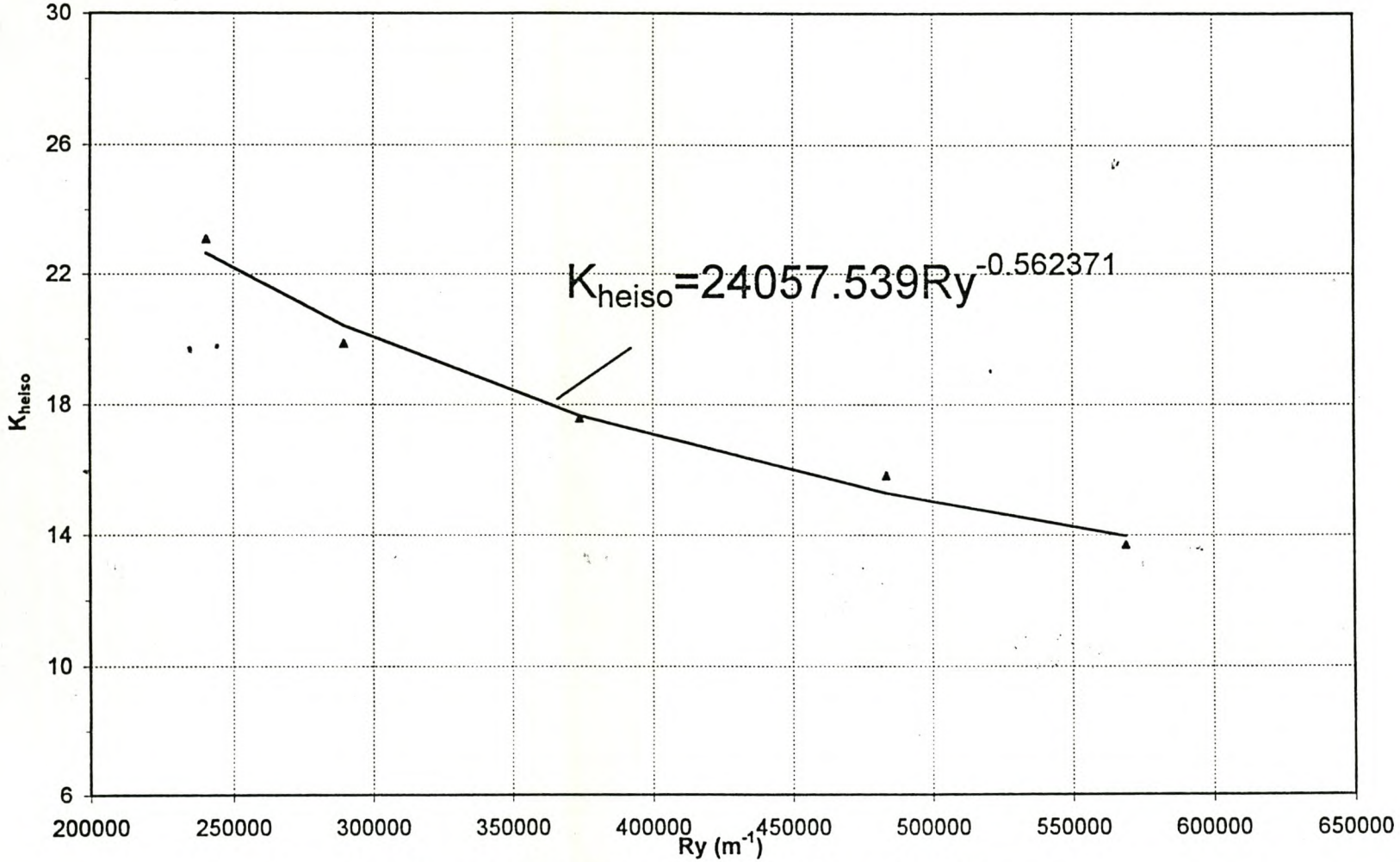



**Figure E.6: Pressure drop coefficient (F- 3.0).**

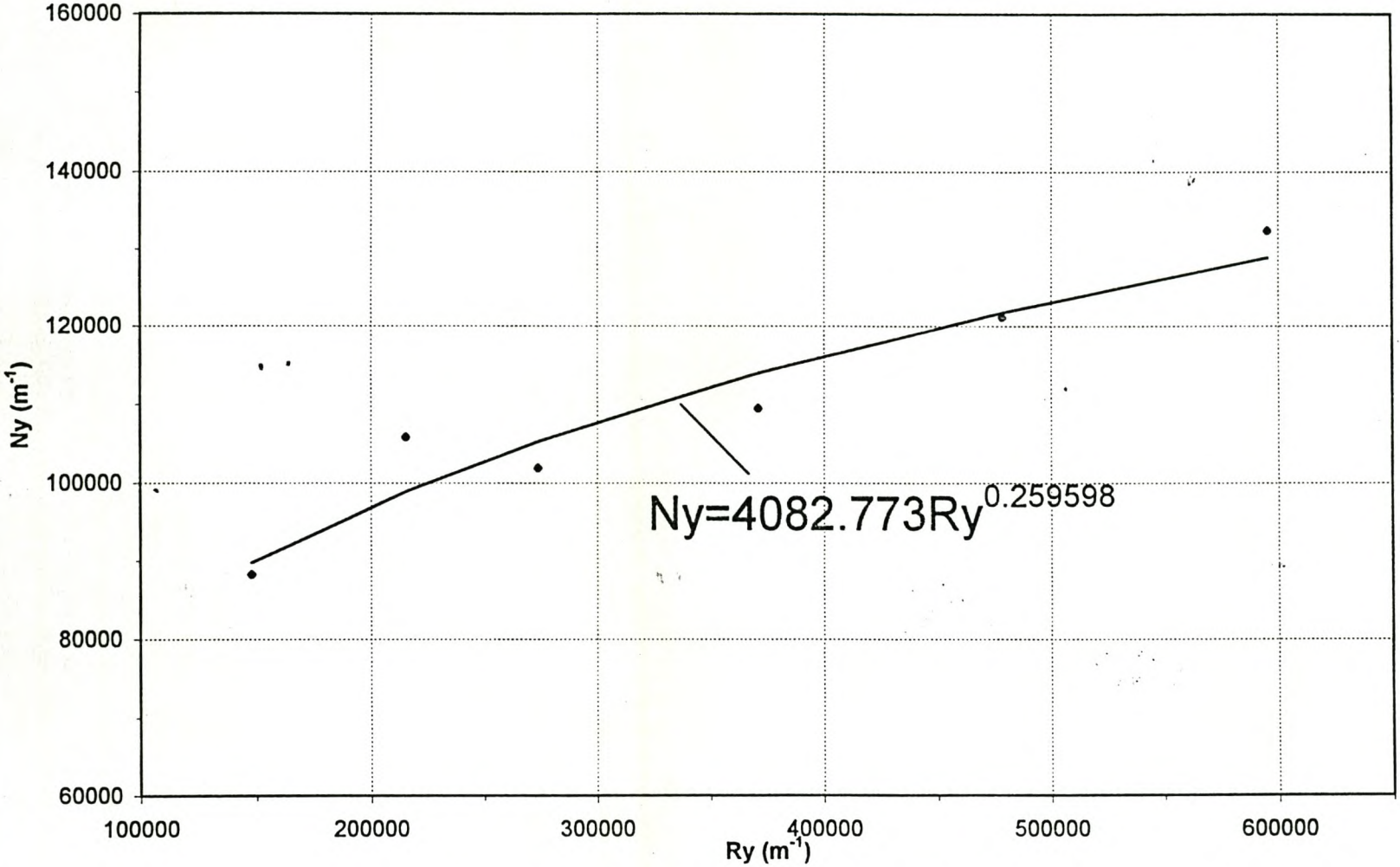



**Figure E.7: Heat transfer parameter (R-2.4G).**

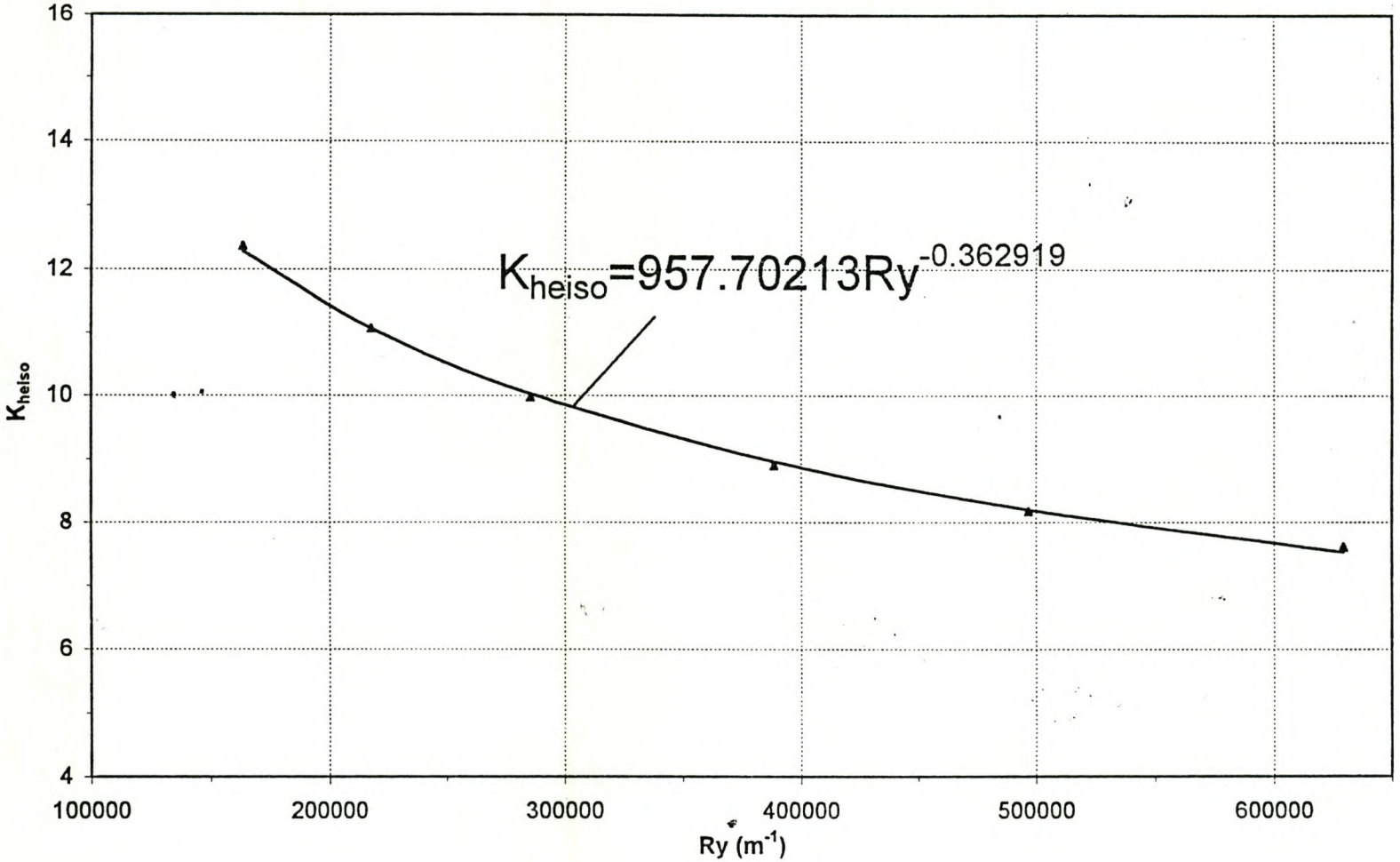



**Figure E.8: Pressure drop coefficient (R-2.4G).**

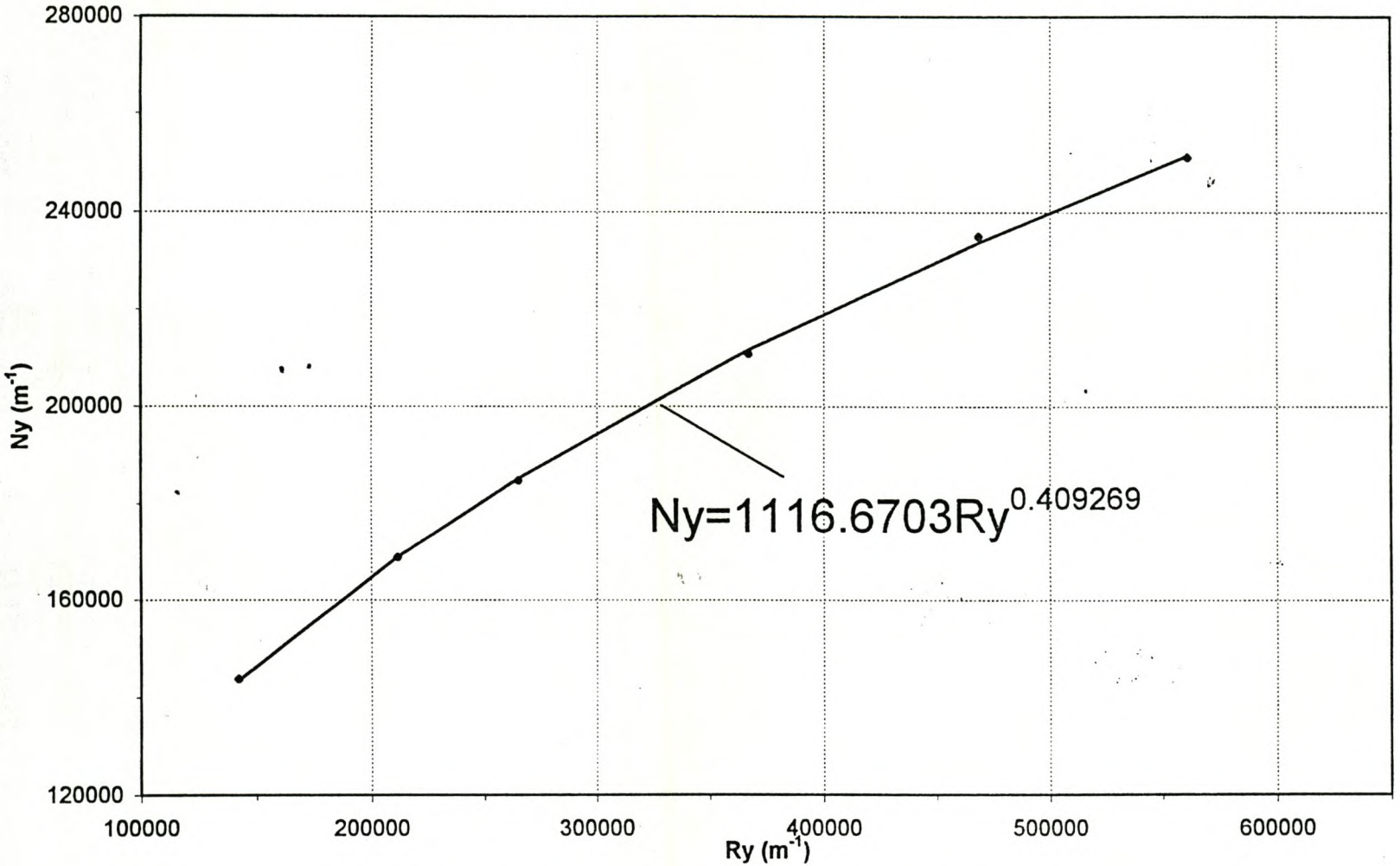



**Figure E.9: Heat transfer parameter (R-2.3D).**

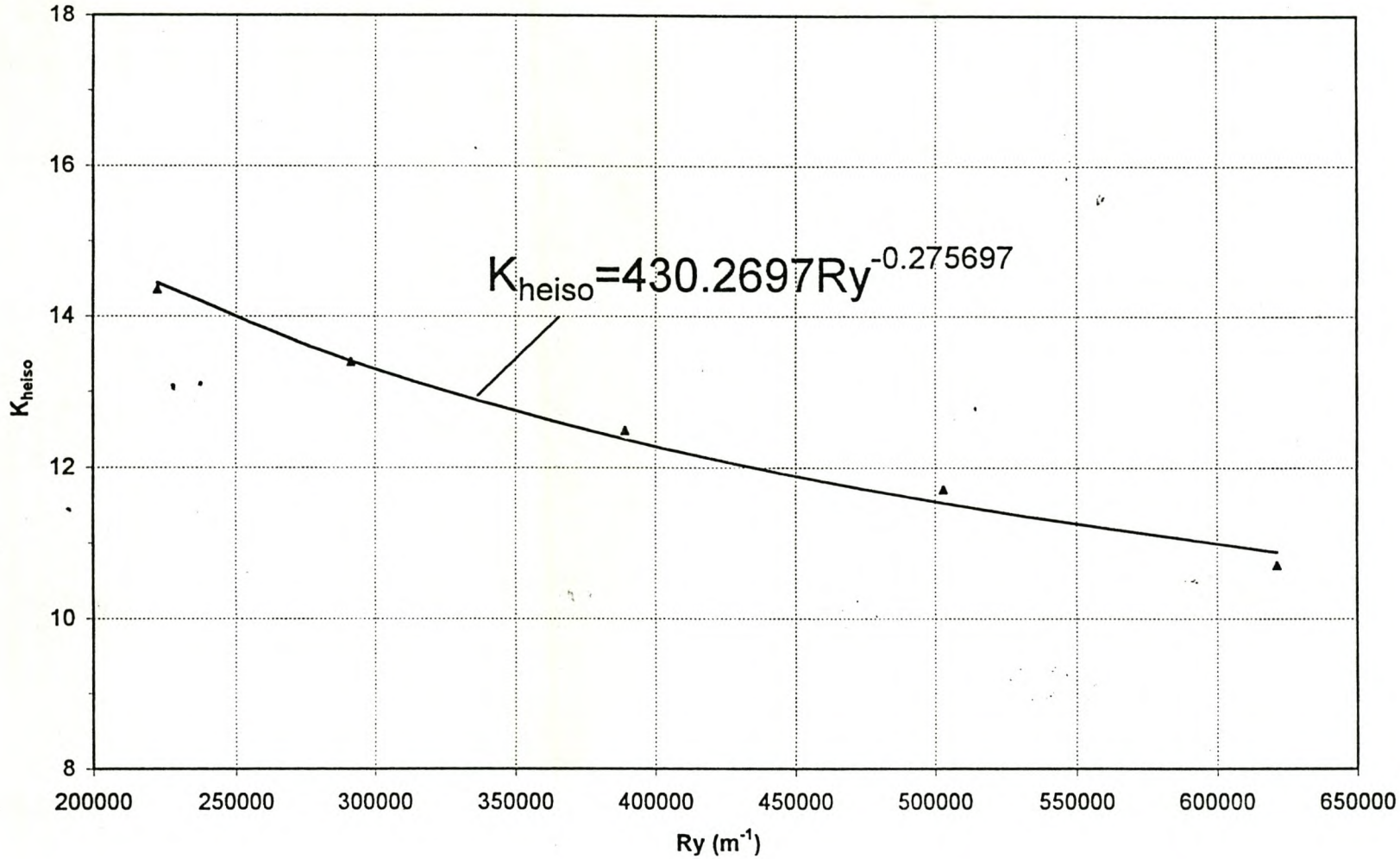



**Figure E.10: Pressure drop coefficient (R-2.3D).**

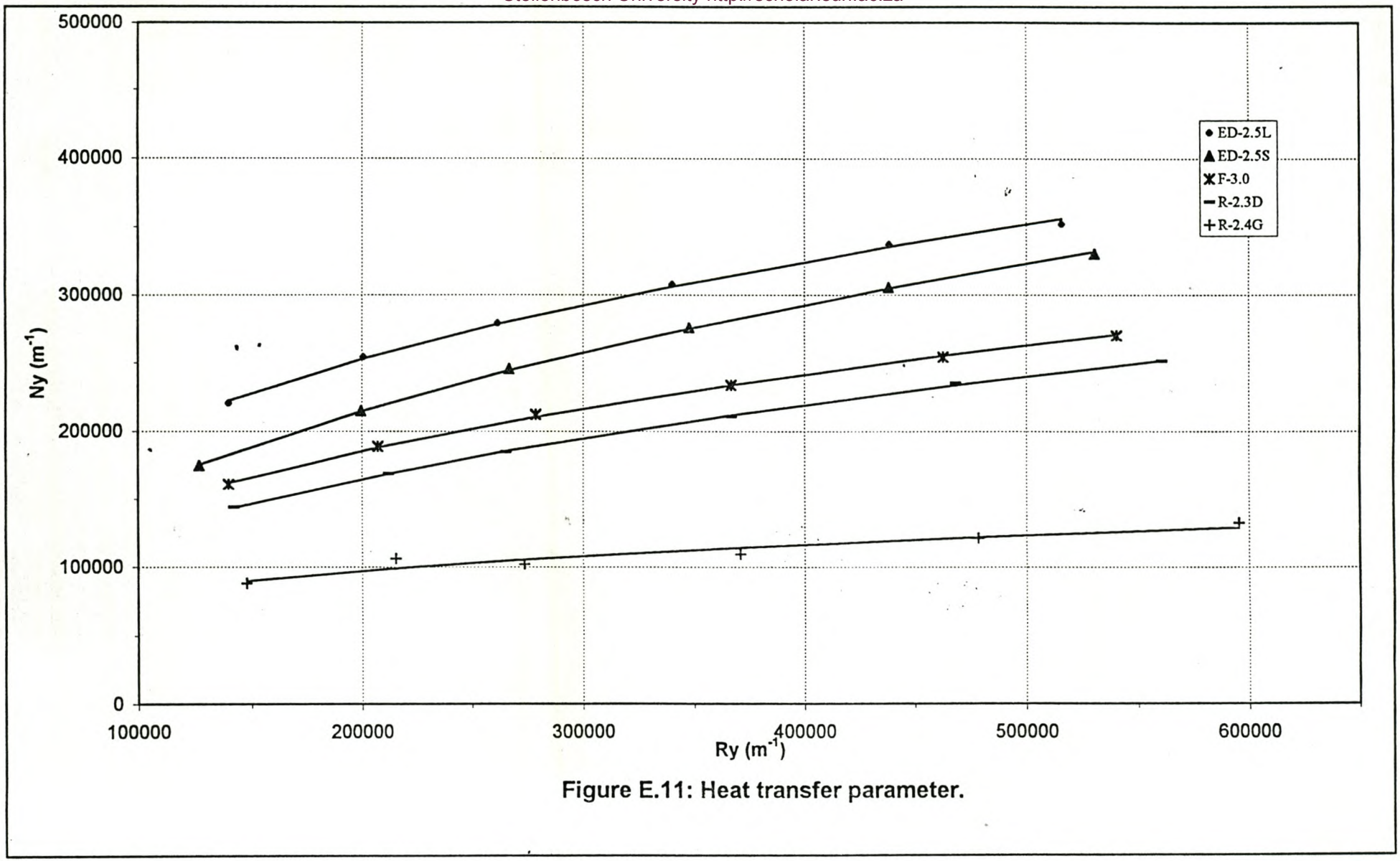



**Figure E.11: Heat transfer parameter.**

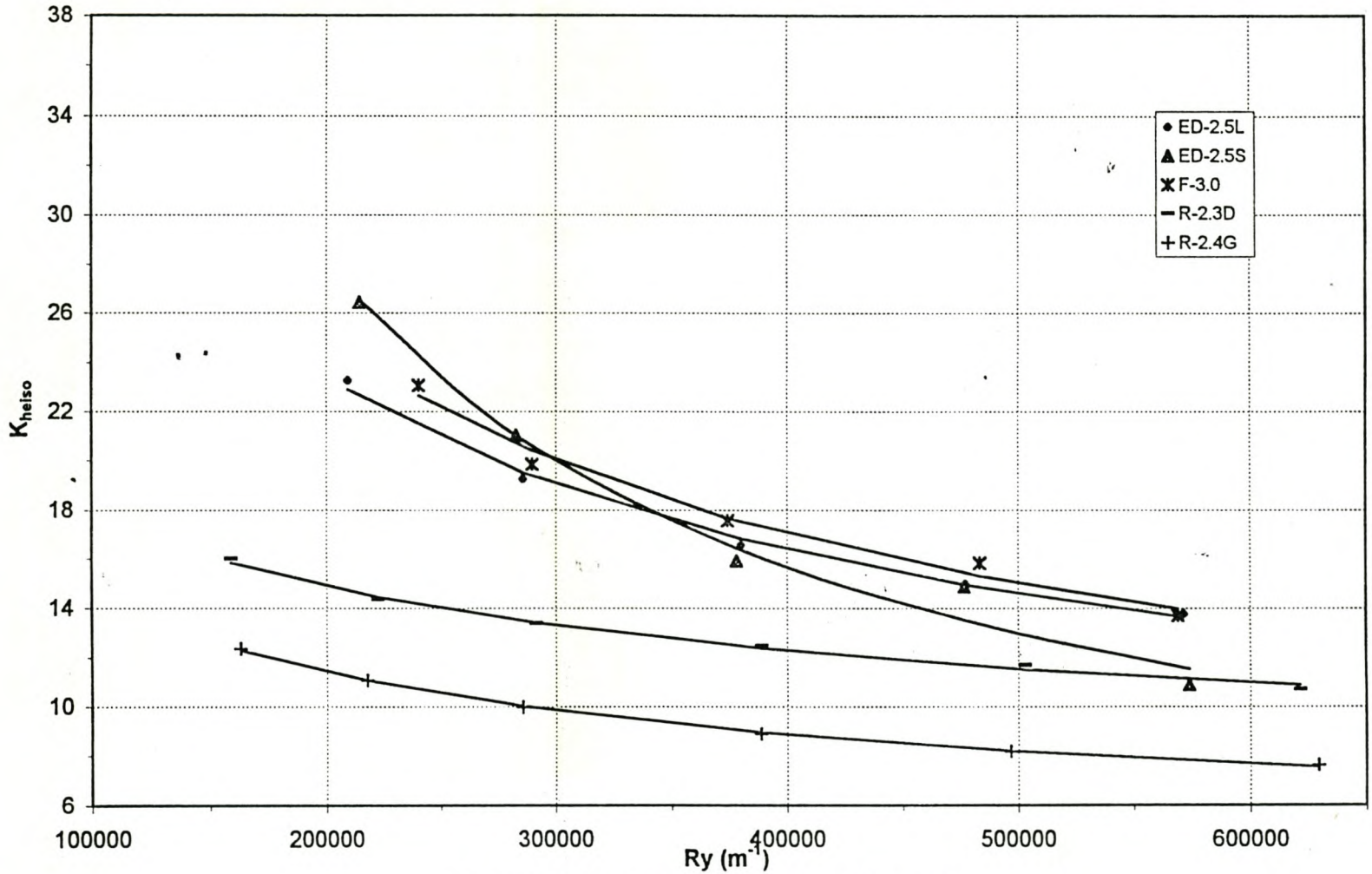



**Figure E.12: Pressure drop coefficient.**

# APPENDIX F

## Sample calculation for a forced-draught air-cooled heat exchanger bay

# Sample calculation for a forced-draught air-cooled heat exchanger bay.

The difference in power consumption between a forced draught and an induced draught air-cooled heat exchanger having the same heat rejection rate is to be evaluated. The two arrangements are depicted schematically in figures 2.1 and 2.2 respectively. The sample calculation is given only for the forced draught arrangement, since the calculation procedure for the induced draught arrangement is similar.

A forced draught air-cooled heat exchanger bank at a petrochemical plant consists of bays having two fans as shown in figure 2.1. Water enters the heat exchanger at a temperature of $T_{wi} = 60$ °C and flows at a rate of $m_w = 100$ kg/s.

Ambient conditions:

| | |
|---|---|
| Air temperature at ground level | $T_{a1} = 28$ °C (301.15K) |
| Atmospheric pressure at ground level | $p_{a1} = 101325$ N/m$^2$ |

The ambient temperature gradient is –0.00975 K/m from ground level and it may be assumed that the air is essentially dry.

Thermophysical properties of water fluid are evaluated at the water inlet temperature of 60 °C (333.15 °C).

| | |
|---|---|
| Specific heat | $c_{pw} = 4183$ J/kgK |
| Thermal conductivity | $k_w = 0.652$ W/mK |
| Dynamic viscosity | $\mu_w = 0.00047$ kg/ms |
| Prandtl number | $Pr_w = 3.015$ |

It may be assumed that these properties do not change significantly with temperature over the cooling range of about 12 °C. The process fluid flows inside finned tubes where its heat transfer coefficient, which includes fouling effects, is specified as $h_w = 31500$ W/m$^2$K.

Finned tube bundle specifications:

The heat exchanger consists of R-2.3D finned tubes (see table E.5.1 in appendix E).

| | |
|---|---|
| Number of bundles | $n_b = 2$ |
| Effective frontal area of one bundle | $A_{fr} = 35.96\ m^2$ |
| Effective length of finned tube | $L_t = 10.203$ m |
| Hydraulic diameter of tube | $d_e = 0.02086$ m |
| Area ratio ($A_c$ = minimum flow area through finned tube bundle) | $\sigma = A_c/A_{fr} = 0.5384$ |
| Number of tube rows | $n_r = 4$ |
| Number of tubes per row | $n_{tr} = 55$ |
| Number of tube passes (process fluid mixes between passes) | $n_p = 2$ |

The experimentally determined characteristic heat transfer parameter (appendix E) for a bundle is given by

$$N_y = 1116.6703\ Ry^{0.409269}$$

This correlation was obtained for very low turbulence intensity in the upstream air.

The isothermal loss coefficient through the bundle is

$$K_{heiso} = 430.2697\ Ry^{-0.275697}$$

Fan installation:

The bay includes two ($n_F = 2$) 4.265m diameter fans having a conical inlet shrouds. Other details are as follows:

| | |
|---|---|
| Fan diameter | $d_F = 4.265$ m |
| Fan casing diameter | $d_c = 4.325$ m |

| | |
|---|---|
| Fan rotational speed | $N_F = 265$ rpm |
| Fan hub diameter | $d_h = 1.524$ m |
| Plenum height | $H_{pl} = 1.7$ m |
| Height of fan platform above ground level | $H_3 = 12.7$ m |
| Fan upstream loss coefficients (based on $A_e$) | $K_{up} = 0.1$ |
| Fan downstream loss coefficient (based on $A_e$) | $K_{do} = 0.15$ |
| Conical shroud inlet loss coefficient | $K_{Fsi} = 0.07$ |
| Heat exchanger support loss coefficient (based on frontal area of heat exchanger) | $K_{ts} = 1.5$ |

The performance characteristics of the 4.265 m diameter fan mounted in a 4.325 m diameter casing with a bellmouth-type inlet and tested according to BS 848 in a type A installation, are specified at a reference air density of $\rho_r = 1$ kg/m$^3$ and $N_{Fr} = 216$ rpm.

Fan static pressure:

$$\Delta p_{Fsr} = 140.2243 + 0.8776\, V_{Fr} - 0.014\, V_{Fr}^2 + 1.5075 \times 10^{-5}\, V_{Fr}^3 \text{ , N/m}^2$$

Fan shaft power consumption

$$P_{Fr} = 31.6268 - 0.9904\, V_{Fr} + 0.019\, V_{Fr}^2 - 1.4427 \times 10^{-4}\, V_{Fr}^3 + 3.7075 \times 10^{-7}\, V_{Fr}^4 \text{ , kW}$$

Losses due to separation at the inlet to the fan platform can be neglected.

## Solution

The problem is solved by an iterative procedure. Energy equations (2.2.1) and (2.2.2) as well as the draught equation (2.2.9) must be satisfied simultaneously. It is found that these equations are indeed satisfied for a total air mass flow rate through the bay of $m_a$ = 317.7933 kg/s. At this flow rate the temperature entering the heat exchanger is $T_{a5}$ = 301.1098 K while its outlet temperature is $T_{a6}$ = 316.7982 K.
Corresponding air densities at these temperatures and a pressure of $p_{a1}$ = 101325 N/m$^2$ are according to equation (2.2.6).

$\rho_{a5} \approx p_{a1} / (RT_{a5}) = 101325 / (287.08 \times 301.1098) = 1.1722 \text{ kg/m}^3$

and similarly,

$\rho_{a6} \approx 101325 / (287.08 \times 316.7982) = 1.1141 \text{ kg/m}^3$

Thermophysical properties of air flowing through the heat exchanger are evaluated at the arithmetic mean temperature i.e. $T_{am} = (T_{a5} + T_{a6}) / 2 = (301.1098 + 316.7982)/2$
$= 308.954$ K.

Mean density,

$\rho_{am} \approx 101325 / (287.08 \times 309.954) = 1.1424 \text{ kg/m}^3$

Specific heat,

$c_{pam} = 1.045356 \times 10^3 - 3.161783 \times 10^{-1} T_{am} + 7.083814 \times 10^{-4} T_{am}^2 - 2.705209 \times 10^{-7} T_{am}^3$

$= 1.045356 \times 10^3 - 3.161783 \times 10^{-1} \times 309.954 + 7.083814 \times 10^{-4} \times 309.954^2$

$- 2.705209 \times 10^{-7} \times 309.954^3 = 1007.355 \text{ J/kgK}$

Dynamic viscosity,

$\mu_{am} = 2.287973 \times 10^{-6} + 6.259793 \times 10^{-8} T_{am} - 3.131956 \times 10^{-11} T_{am}^2 + 8.15038 \times 10^{-15} T_{am}^3$

$= 2.287973 \times 10^{-6} + 6.259793 \times 10^{-8} \times 309.954 - 3.131956 \times 10^{-11} \times$

$309.954^2 + 8.15038 \times 10^{-15} \times 309.954^3 = 1.8924 \times 10^{-5} \text{ kg/m}$

Thermal conductivity,

$k_{am} = - 4.937787 \times 10^{-4} + 1.018087 \times 10^{-4} T_{am} - 4.627937 \times 10^{-8} T_{am}^2$

$$+ \ 1.250603 \times 10^{-11} T_{am}^{3}$$

$$= - 4.937787 \times 10^{-4} + 1.018087 \times 10^{-4} \times 309.954 - 4.627937 \times 10^{-8} \times 309.954^{2}$$

$$+ 1.250603 \times 10^{-11} \times 309.954^{3} = 2.6989 \times 10^{-2} \text{ W/mK}$$

Prandtl number,

$$Pr_{am} = \mu_{am} c_{pam} / k_{am} = 1.8924 \times 10^{-5} \times 1007.355 / 2.6989 \times 10^{-2} = 0.70633$$

The total effective frontal area of the heat exchanger in the bay is given by

$$A_{frtot} = n_b A_{fr} = 2 \times 35.96 = 71.92 \text{ m}^2$$

Find the characteristic flow parameter

$$Ry = m_a / (\mu_{am} A_{frtot}) = 317.7933 / (1.8924 \times 10^{-5} \times 71.92) = 233497.437 \text{ m}^{-1}$$

According to the specified relation, the corresponding heat transfer parameter is

$$Ny = 1116.6703 \times 233497.437^{0.409269} = 175790.912 \text{ m}^{-1}$$

It follows from equation (3.4.6) that

$$h_{ae} A_a = Ny k_{am} A_{frtot} Pr_{am}^{0.333} = 175790.912 \times 2.6989 \times 10^{-2} \times 71.92 \times 0.70633^{0.333}$$

$$= 303915.213 \text{ W/K}$$

The total inside surface area of the tubes that is exposed to the water stream can be expressed as

$$A_w = n_{tr} n_r L_t n_b \pi d_e = 55 \times 4 \times 10.203 \times 2 \times \pi \times 0.02086 = 294.201 \text{ m}^2$$

With the above values find the overall heat transfer coefficient.

$UA = (1/h_{ae}A_a + 1/h_wA_w)^{-1} = (1 / 303915.213 + 1/(31500 \times 294.201))^{-1} = 294265.012$ W/K

To find the number of transfer units, it is noted that $C_{max} = m_wc_{pw} = 100 \times 4183 = 418300$ W/K > $C_{min} = m_ac_{pam} = 317.7933 \times 1007.355 = 320130.6697$ W/K such that

$NTU = UA/C_{min} = 294265.012 / 320130.6697 = 0.9192$

There are two tube-side passes, with each pass having

$NTU_p = N_p = NTU / 2 = 0.9192 / 2 = 0.4596$

Furthermore, the heat capacity ratio

$C = C_{min} / C_{max} = 320130.6697 / 418300 = 0.76531$

If both the air stream and the process fluid are unmixed in each pass, then the effectiveness per pass is given as [84KA1],

$e_p = 1- \exp[N_p^{0.22}\{\exp(-CN_p^{0.78}) - 1\} / C]$

$= 1- \exp[0.4596^{0.22}\{\exp(-0.76531 \times 0.4596^{0.78}) - 1\} / 0.76531] = 0.31323$

With this value it follows that the effectiveness of the entire heat exchanger, where mixing of the process fluid occurs between passes, is

$e = [\{(1 - e_pC) / (1 - e_p)\}^{np} - 1] / [\{(1 - e_pC) / (1 - e_p)\}^{np} - C]$

$= [\{(1 - 0.31323 \times 0.76531) / (1 - 0.31323)\}^2 - 1] / [\{(1 - 0.31323 \times 0.76531) /$

$(1 - 0.31323)\}^2 - 0.76531] = 0.49006$

According to equation (2.2.2) the rate of heat transfer is thus

$Q = eC_{min}(T_{wi} - T_{a5}) = 0.49006 \times 320130.6697[ 60 - (301.1098 - 273.15)) = 5.0265$ MW

The heat transfer rate can also be determined from equation (2.2.1) i.e.

$$Q = m_a c_{pam}(T_{a6} - T_{a5}) = 317.7933 \times 1007.355\ (316.7982 - 301.1098) = 5.0223\ \text{MW}$$

These two values for the heat transfer rate are in excellent agreement.

To determine whether the draught equation is satisfied, the fan performance has to be evaluated.

The approximate air temperature at the fan inlet is

$$T_{a3} \approx T_{a1} - 0.00975H_3 = 301.15 - 0.00975 \times 12.7 = 301.026\ \text{K}$$

At this temperature and the specified atmospheric pressure at ground level, the approximate air density is given by equation (2.2.6)

$$\rho_{a3} \approx 101325 / (287.08 \times 301.026) = 1.172\ \text{kg/m}^3$$

and the specific heat is

$$c_{pa3} = 1.045356 \times 10^3 - 3.161783 \times 10^{-1}\, T_{am} + 7.083814 \times 10^{-4}\, T_{am}^2 - 2.705209 \times 10^{-7}\, T_{am}^3$$

$$= 1.045356 \times 10^3 - 3.161783 \times 10^{-1} \times 301.026 + 7.083814 \times 10^{-4} \times 301.026^2$$

$$- 2.705209 \times 10^{-7} \times 301.026^3 = 1006.99\ \text{J/kgK}$$

Actual air volume flow rate through each fan is

$$V = m_a / (n_F \rho_{a3}) = 317.7933 / (2 \times 1.172) = 135.577\ \text{m}^3/\text{s}$$

Since the actual density and rotational speed of the fan are not the same as the reference conditions for which fan performance characteristics were specified, the relevant fan laws [98KR1] are employed.

The reference air volume flow rate is

$V_{Fr} = V(N_{Fr}/N_F) = 135.577 \times 216/265 = 110.508\ m^3/s$

At this flow rate the reference fan static pressure rise is given by

$\Delta p_{Fr} = 140.2243 + 0.8776 \times 110.508 - 0.014 \times 110.508^2 + 1.5075 \times 10^{-5}$

$\times 110.508^3 = 86.582\ N/m^2$

The actual change in fan static pressure is

$\Delta p_{Fs} = \Delta p_{Fr}(N_F / N_{Fr})^2(\rho_{a3} / \rho_r) = 86.582(265 / 216)^2(1.172 / 1) = 152.735\ N/m^2$

At the reference condition, the fan shaft power is

$P_{Fr} = 31.6268 - 0.9904 \times 110.508 + 0.019 \times 110.508^2 - 1.4427 \times 10^{-4} \times 110.508^3$

$+ 3.7075 \times 10^{-7} \times 110.508^4 = 14.8032\ kW$

The actual fan shaft power follows is

$P_F = P_{Fr}(N_F / N_{Fr})^3(\rho_{a3} / \rho_r) = 14.8032(265 / 216)^3(1.172 / 1) = 32.0375\ kW$

With this value it is possible to determine the approximate temperature of the air immediately upstream of the heat exchanger [98KR1]

$T_{a5} \approx T_{a1} + P_F / (m_a c_{pa3}) - 0.00975(H_3 + H_{pl})$

$= 301.15 + 32037.5 / (317.7933 \times 1006.99) - 0.00975 \times (12.7 + 1.7) = 301.1097$

This value is essentially as given initially.

The fan unit pressure rise coefficient is

$$K_{Fs} = 2\Delta p_{Fs}\rho_{a3} / (m_a/(n_F A_c))^2 = 2 \times 152.735 \times 1.172 / [317.7933 \times 2 / (\pi \times 4.325^2)]^2$$

$$= 3.0605$$

It follows from equation (3.4.17) that for non-isothermal flow the heat exchanger loss coefficient in this case is given by

$$K_{he} = 430.2697\ Ry^{-0.275697} + 2(\rho_{a5} - \rho_{a6}) / \sigma^2(\rho_{a5} + \rho_{a5})$$

$$= 430.2697(233497.437)^{-0.275697} + 2 \times (1.1722 - 1.1141) / \{0.5384^2 \times (1.1722$$

$$+ 1.1141)\} = 14.4222$$

The recommended plenum recovery factor $K_{rec} = 0.3$ for $15 \leq K_{he} \leq 21$. Furthermore, for $H_{pl} / d_c = 1.7 / 4.325 = 0.393 > 0.3$ the corresponding heat exchanger outlet kinetic energy correction factor is

$$\alpha_{e6} = 1.6 - 0.48\ A_c / A_{fr} - 0.012K_{he} = 1.6 - 0.48 \times \pi \times 4.325^2 / (4 \times 35.96) - 0.012$$

$$\times 14.4222 = 1.2308$$

The upstream and downstream loss coefficients, $K_{up}$ and $K_{do}$, are based on the effective fan area

$$A_e = A_c - A_h = \pi(d_c^2 - d_h^2) / 4 = \pi (4.325^2 - 1.524^2) / 4 = 12.8672\ m^2$$

Since no windwall is provided, $H_6 = H_7$ and the left-hand side of the draught equation (2.2.9) is equal to zero. The terms on the right-hand side of the equation give

$$K_{ts} (m_a / A_{frtot})^2 / (2\rho_{a3}) + K_{Fsi} (m_a / (2A_c))^2 / (2\rho_{a3}) ) + K_{up} (m_a / (2A_e))^2 / (2\rho_{a3})$$

$$- ( K_{rec} + K_{Fs} ) (m_a / (2A_c))^2 / (2\rho_{a3}) + K_{do} (m_a / (2A_e))^2 / (2\rho_{a3} ) + K_{he} (m_a / A_{frtot})^2 / (2\rho_{am})$$

$+ \alpha_{e6} (m_a / A_{frtot})^2 / (2\rho_{a6})$

$= 1.5 \times (317.7933 / 71.92)^2 / (2 \times 1.172) + (0.07 - 3.0605 - 0.3) \times (317.7933 \times 2 /$

$(\pi \times 4.325^2))^2 / (2 \times 1.172) + (0.1 + 0.15) \times (317.7933 / (2 \times 12.8672))^2 / (2 \times 1.172) +$

$14.4222 \times (317.7933 / 71.92)^2 / (2 \times 1.1424) + 1.2308(317.7933 / 71.92)^2 / (2 \times 1.1141)$

$= -1.4232 \text{ N/m}^2$

This value is close to zero and therefore the draught equation is satisfied.

Finally, table F.1 below shows comparison in performance between a forced as well as an induced draught air-cooled heat exchanger.

Table F.1: Performance comparison of a forced and an induced draught ACHE.

| | **Forced draught ACHE** | **Induced draught ACHE** |
|---|---|---|
| Heat rejected, $Q_{rem}$ (MW) | 5.022 | 5.022 |
| Fan shaft power, $P_F$ (kW) | 32.0583 | 34.3135 |
| Air mass flow rate, $m_a$ (kg/s) | 317.7933 | 316.799 |
| Fan rotational speed, $N_F$ (rpm) | 265 | 276 |
| Air outlet temperature, °K | 316.7982 | 316.799 |